COMBUSTION

FEB 23 1939

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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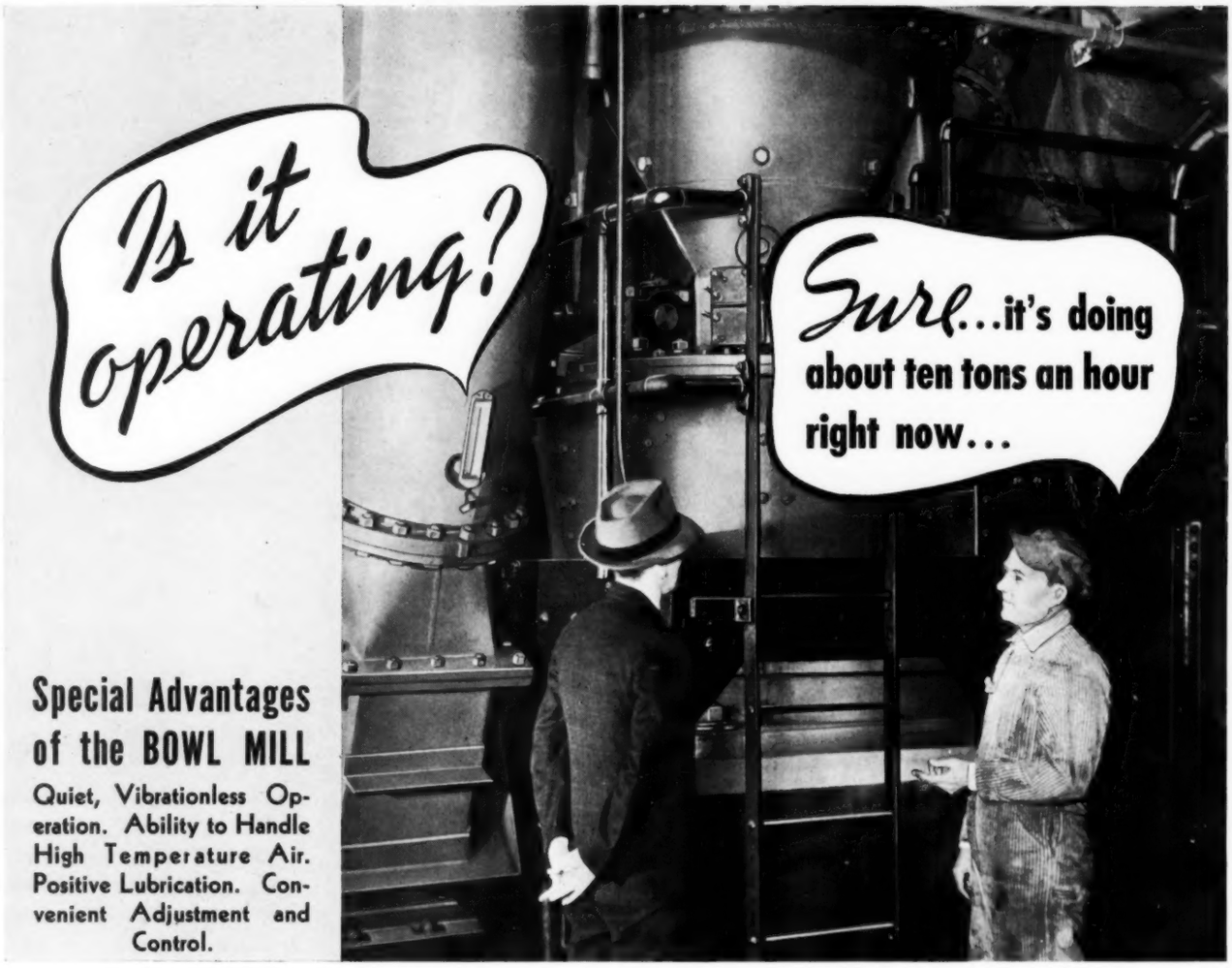
Is America Ready for Forced-Circulation Boilers?

Deaerators and Their Applications

Cylinder Oil in Condensed Steam

Efficiency Comparisons of Turbine-Generators and Generating Stations



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COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

VOLUME TEN NUMBER EIGHT

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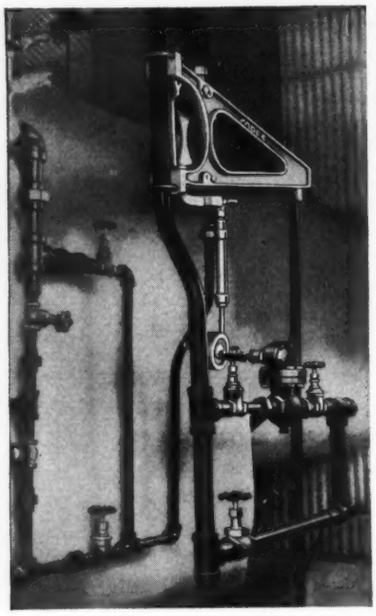

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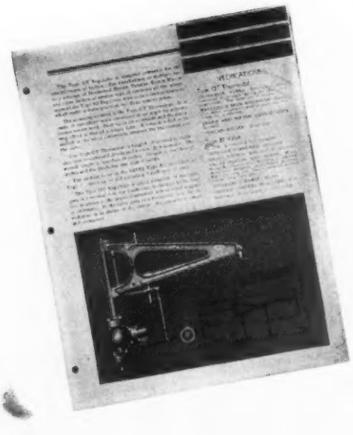

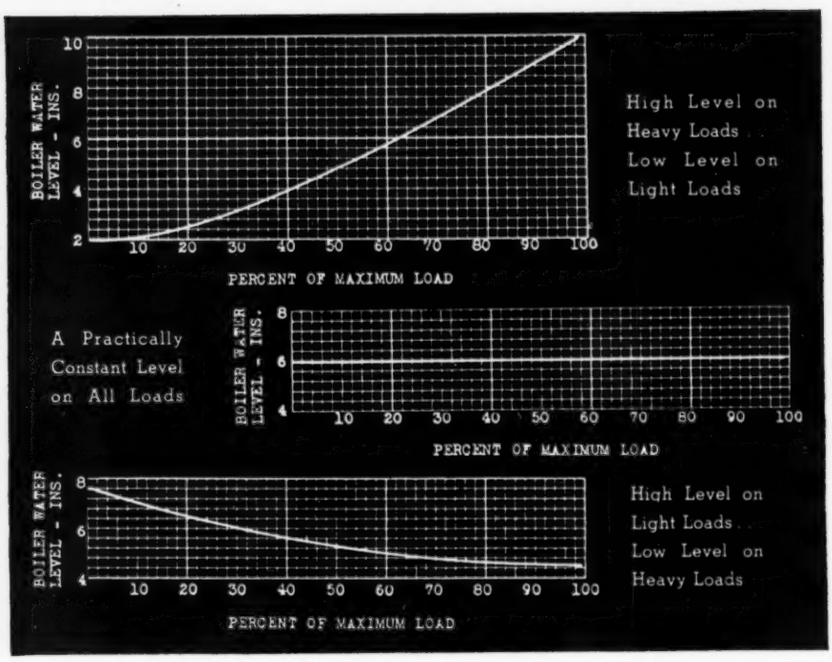

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EDITORIAL

Professional Engineering versus Unionism

During recent years the status of professional engineers has been enhanced by the adoption of laws in most States providing for licensing and registration upon meeting the required qualifications. Public recognition has been further advanced by the efforts of the Society of Professional Engineers through its National, State and Local bodies. The ground gained must be zealously guarded and it is natural that anything which savors of a departure from the high ideals set up should be critically regarded.

In this connection, an incident has lately come to the attention of the officers of the New York State Society of Professional Engineers in which a member initiated the formation of an organization designated as "Supervising Professional Engineers Union," under charter from a local of one of the two national labor organizations. Accompanying this, a blind advertisement for engineers at attractive salaries, was inserted in one of the leading New York papers and those who responded are reported to have been solicited to join the union at a high initiation fee. It is said that the prospective positions had to do with inspection of certain building operations in New York City and were to be correlated with the activities of the local building trades union.

This led the President of the New York State Society of Professional Engineers to issue the following announcement:

"Engineers and other professional men do not belong in labor unions and should not be solicited, nor encouraged to join. Unionization is inconsistent with the professional spirit and attitude, which places the highest service to the public above personal gain. The State Societies of Professional Engineers and their National Society are interested in improving the economic and professional status of engineers, but believe this should be accomplished through education and public appreciation; not through methods of unionization. We recognize that unionization has its place, but that place is not in the learned professions. Professional men occupy a position of public trust, between capital and labor, and unionization is inconsistent with the highest discharge of that trust."

Although it is true that the constitution of the Society of Professional Engineers does not prevent a member from joining a union, both the State and National bodies have previously gone on record as fundamentally opposed to the idea of unionization for professional men. This is not because of antipathy toward labor unions, but largely for the reasons stated and, further, because it is highly desirable that such men be unfettered in the exercise of their duties which often involve relations between employers and labor. In many cases professional engineers are themselves employers of labor and, in the event of their unionization, it is conceivable that most confusing situations might arise. Such practical considerations outweigh individual opinions. Moreover, it is incontrovertible that unionism exercises a certain leveling effect which is inconsistent with the evaluation of professional service. Finally, any move toward trade unionization of engineers is very likely to undo years of effort to increase public recognition of engineering as a profession.

Passamaquoddy Again

Unabashed by previous rebuffs, the President has recommended that Congress appropriate funds to resume work on the Passamaquoddy project to harness the tides of the Bay of Fundy for power production. The estimated cost of this development is around thirty-six million dollars, although it is proposed, for the present, to confine the work to borings and a partial development on the American side.

Even though such an initial development as is proposed be regarded as an experimental installation, it is likely to lead to further substantial commitments.

It will be recalled that this project was begun several years ago under a lump sum appropriation for W.P.A. work. After several millions had been spent, principally for housing the construction force and in preliminary surveys, Congress became convinced of its uneconomic value and in 1936 refused further appropriations to carry on the work. Since then the numerous buildings that were erected have been utilized for activities of the National Youth Administration.

The idea of harnessing these tides is not recent. It was proposed a number of years ago but did not survive critical engineering analysis. Had the President consulted competent engineering opinion in the beginning it is possible that the initial investment might have been saved. The plan, while perhaps physically feasible, has been pronounced economically unsound in view of the ability to produce power from fuel at much less cost and the fact that adequate power is already available in the region to be served. The plea that revival of work on this project will help alleviate unemployment in Northern Maine has resulted in some local support for the proposal, but it is conceivable that other, more rational, means could be found to ease the local situation.

Any attempt to associate the project with power for national defense is absurd.

Is America Ready for Forced-Circulation Boilers?

By ALFRED R. SMITH

Combustion Engineering Company, Inc.

The following discussion points out some of the reasons why forced-circulation boilers have thus far made greater headway in Europe than in the United States. However, despite the fact that Germany leads in applications of this type, over half of her high-pressure installations employ natural circulation. The comments do not apply to marine, railway or special services which under certain conditions may favor forced circulation; nor is any attempt made to predict what the future holds for forced circulation in this country inasmuch as it is recognized that changed economic conditions often carry equal weight with technical considerations. Representative types of European forced-circulation boilers are briefly described.

WHEN developments in forced circulation abroad are viewed and these advances studied the first question that comes to mind is, "Should we be doing this task in this fashion?" The answer depends upon the degree of similarity of objectives, as dictated by attendant conditions here and abroad. These directing circumstances must be studied as carefully as the technical details if a clear understanding of the problem is to be had. The scope of the present discussion will be confined to the field of stationary steam practice for large scale power generation under average conditions, without reference to boilers for the production of steam in the marine field, in the realm of railway motive power, in laboratory investigations, or in special process applications, as it is recognized that these may involve conditions and problems requiring special consideration. The successful existence of such boilers is not ignored, however, when the teachings to be derived exert an influence on the stationary field.

Extent of Forced Flow in Natural-Circulation Units

In many modern designs of so-called natural-circulation steam generating units, over two-thirds of the tube surface actually operates under forced flow and absorbs about one-third of the heat delivered from the fuel to the feedwater. This applies to the economizer and superheater surfaces. By natural-circulation surface reference is made to the furnace walls, the boiler convection surface and to any gas exposed risers. Alteration of present designs for full forced circulation would involve changes in the method of supplying water to the furnace walls and to the boiler convection surface. The latter is a relatively small part of the total, being in the average large high-pressure unit about a sixth of the total heat absorption surface. Of the two-thirds of the total heat which is absorbed by natural-circulation surfaces, a large part is accounted for by the furnace walls. Since the boiler proper absorbs about one-sixth of the total, it follows that the furnace walls pick up about one-half. Changing to forced circulation then becomes essentially a question as to whether the furnace walls shall be supplied with mechanically motivated liquid. Such is the extent, on this score, of the divergence of practice here and abroad.

Drums and Tubes

In some designs of forced-circulation boilers the drums are omitted and in most cases the tubes are small. How advantageous are such alterations in basic design? The answer abroad is definitely associated with the cost and availability of steel and the cost of labor; but in this country the plentiful supply of steel of the required properties may be contrasted with conditions abroad where this commodity is at present at a premium, for obvious and unfortunate reasons. Some designs of once-through-flow boilers have achieved a weight reduction of two-thirds of that which would be required by the conventional design for similar duty. Hence, for the express objective of weight reduction, the achievement is commendable. On the other hand, in America the use of steel for industrial purposes is not viewed with alarm; indeed, the opposite reaction is entailed. Economically, the employment of greater quantities of materials rather than lesser amounts is sound. Such use cannot even be said to be a drain on our natural ore resources because much of the iron so used eventually returns to commerce as scrap and because our ore resources are so vast as to render such withdrawals trivial. The sum of the cost of material and labor is the true criterion for judgment.

Also, decentralization of power generating equipment for reasons of military expediency, such as is now being urged abroad, favors development of types which can be constructed in small sizes.

Opposing these advantages, certain attendant disadvantages may be pointed out. In those designs which eliminate the drum the water storage capacity of the unit is decreased and it requires closer and more responsive control. If the required time for response of

the control is taken as unity for the natural-circulation design, to maintain an equivalent degree of stability of output, the forced-circulation boiler with some drum capacity must have a control which will act in one-third of the time, while the once-through-flow boiler must be controlled in one-sixth of the time. Also, the drum affords a point of accumulation from which scale may be eliminated by blowdown. In drumless designs other expedients must be employed for elimination of salts from the system.

The use of small tubes may be viewed in the light of relative labor costs here and abroad. The cost of providing surface in a unit is the sum of the cost of material and services. The principal part of the expense for services is the cost of labor for fabrication and for installation. In this country the unit rate for the former is smaller, for which reason field labor is usually avoided as much as is practically possible with the result that designs involve mainly the use of shop labor. Many of the forced-circulation boilers necessitate field welding of a great many small tubes. Moreover, requirements for field welds are more stringent here than they are abroad.

On the other hand, where both high temperatures and very high pressures are employed, the use of large tubes of heavy wall thickness may involve stresses such as to make it desirable to consider the use of small tubes of reduced wall thickness. Small tubes in natural-circulation boilers involve certain elements of design and cost that are non-existent with large tubes at lower pressures.

A further requirement that is sometimes advanced against the forced-circulation boiler is the power demanded by the circulating pump over and above feed requirements. The differential is brought about by the greater resistance to flow through small tubes and the lengthened path for such flow, whether the unit be once-through or recirculating. This difference may or may not be of importance, depending on the design. That it is not always excessive is indicated by the recently published tests of the La Mont boiler installed in the plant of G. & J. Weir, Ltd., near Glasgow, where the power absorbed by the circulating pump accounted for only 0.55 per cent of the output of the unit, despite the employment of restricting orifices to assure greater equality of distribution through the parallel circuits of the boiler.

While the power consumed by the circulating pump must be debited against the efficiency of the unit, this may be considered as offset by affording flexibility in design and arrangement of surfaces such as will lessen circulation difficulties.

Circulation and Pressure

There is ample evidence to support the tenet that natural circulation is adequate for pressures up to 1500 lb per sq in., and strong evidence for the belief that natural circulation may also be adequate for pressures up to 2500 lb.[1]

There has been little demand in the past for boiler pressures to exceed 1500 lb. In meeting an inquiry for a unit to operate at 2000 to 2500 lb, it would be sound engineering to investigate the latitude in arrangement of surfaces, diameter of tubes and reduced head room requirements of a forced-circulation boiler. Such pressures may prove desirable for topping existing stations now operating at around 450 to 500 lb. By and large, however, it does not seem unreasonable to assume that for some time to come boilers for pressures above 1500 lb will be few.

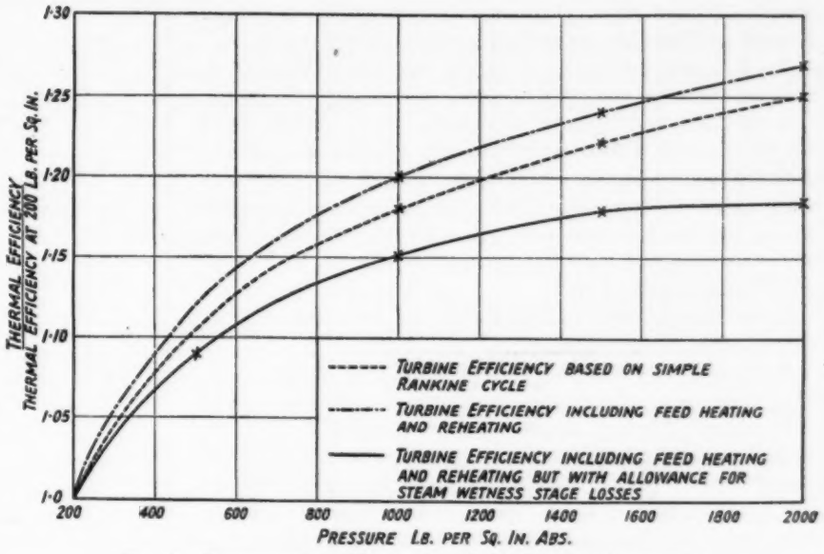

Fig. 1—Comparative efficiencies plotted against pressure

If the pressure for which natural circulation is adequate is passed, it becomes necessary to show tangible reasons for the adoption of higher pressures. While the basic cycle shows certain thermal advantages in the range prohibited for natural circulation, these advantages diminish as other factors are introduced such as lesser stage efficiency due to increased wetness of the steam, and added power for driving the feed and circulation pumps. The curves prepared by Prof. A. L. Mellanby[2] are here reproduced in Fig. 1 as being of considerable interest in this respect. It will be noted that the efficiency improves by only about 1 per cent in going from 1500 to 2000 lb. This does not take into account power for circulating or for additional feed pump resistances introduced by operation under forced circulation.

If the highest range of pressures is to be widely invaded, metallurgy must furnish materials which will enable the satisfactory operation of superheaters beyond the present limit of 900 to 1000 F. As is so universally usual in engineering, the added increments of efficiency are often obtained at increasingly higher costs.

European Use of Forced Circulation

The importance of an apparatus is often judged by its extent of use. There are installed in Europe for stationary use several hundred forced-circulation boilers, the majority designed for pressures above 1130 lb per sq in. In Germany are to be found, according to recent published reports, a total of 189 units of various types for high-pressure service, of which about half are of conventional design as we know it, about three-eighths

[1] See "Circulation in Boiler Tubes," by Kurt Toensfeldt, COMBUSTION, Aug. 1937, vol. 9, no. 2.

[2] See *Power and Works Engineer*, January 1938. "Why Exceed 1000 lb per sq in.?" by Prof. A. L. Mellanby.

employ forced circulation and the balance are dual-circuit natural-circulation units of the Schmidt type.

In Russia, the ratio of recent forced-circulation to natural-circulation applications is probably higher. The Soviet government is reported to have standardized on once-through-flow boilers of the Ramsin design and its 1937 power expansion program called for twenty-six such units. One boiler of this type of large size (440,000 lb per hr) has been in operation for well over two years, and two large Loeffler units have been in service in Moscow for several years.

Forced-circulation boilers have also been installed in other Continental countries.

England has adopted high pressures reluctantly; hence the number of such installations is limited. The Brimsdown Station employs two 210,000-lb per hr forced-circulation boilers of the Loeffler type for generating steam at around 1900 lb pressure, and a 350,000-lb per hr La Mont unit for 375 lb steam pressure is being installed at the Deptford West Station. A high-pressure La Mont boiler of 50,000 lb per hr capacity has lately gone into service in a British industrial plant. On the other hand, the 1350-lb extension to Battersea Station will house two 500,000 lb per hr units of the natural-circulation type. Other installations for high pressure are applications outside of the scope of the present discussion.

Forced-Circulation Boiler Types

The principal designs of forced-circulation boilers will be considered against the background furnished by the foregoing comments. It is to be remembered that the present discussion is limited to large boilers in the central station field. An entirely different background would be required for consideration of similar functional designs in the marine field. Railway motive power, in so far as it imposes a still different set of conditions, would require other measures for judgment. The descriptions of the more frequently used designs will be confined, with a single exception, to those identified as La Mont, Loeffler, Sulzer, Benson and Ramsin. All of these have been used in relatively large sizes at high pressure for central-station application. The Schmidt boiler is normally a natural-circulation type. However, it will be included since it has been employed to considerable extent in the high-pressure field, and since it is of interest to compare its characteristics with American practice. The Velox boiler is not included as it is not strictly a forced-circulation type and because it employs pressure combustion and is thus limited to the burning of liquid or gaseous fuels. It is in a class by itself.

Apart from the general arrangement of surfaces and their interrelation, it is of importance to consider designs on the basis of methods of control and means for salt elimination. Much of the novelty that exists in the field of forced or assisted circulation has to do with control methods.

The Benson Boiler

This design is of special interest because of two features not found on the earlier examples of drumless forced-circulation, once-through-flow boilers. These are: first, the inclusion of a salt deposit zone located in a relatively cool region following the superheater; and second, the use of surface before and after the conversion zone, or transition section, for the purpose of control. This arrangement of the circuit will be apparent by reference to Fig. 2.

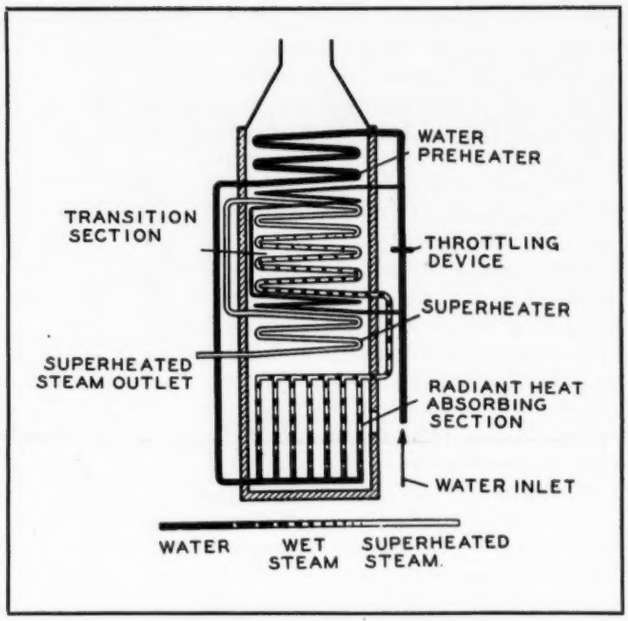

Fig. 2—Circuit of Benson boiler

If the transition from water to steam were to be completed in a zone of high temperature, the depositing salts would insulate the tube to an extent that would bring about overheating and rupture. It is said that the point of transition can be maintained in the bank above the superheater from which the salt is automatically eliminated during starting and shutting down. Where the boiler is operated continuously over long periods, arrangements can be made to flush out salt deposits successively from the deposition zone by suitable subdivision and valving arrangements.

Control of the Benson boiler is effected through temperature changes in a few coils of tubes through which a small portion of the feedwater is diverted. This water is not evaporated but merely heated. The water is divided by means of a throttling device so that it is a definite proportion of the total feed, about one one-thousandth part. Changes in temperature reflect the heating ability of the gases. As the temperature falls, fuel and air feed are immediately increased, giving an indication of the correlation of firing conditions with demand more promptly than would be afforded by the superheated steam leaving the unit, owing to the heat content of the tubes. The coil arrangement acts as a corrective supplement to load-controlled regulation adjustments of fuel, air and water.

The more recent designs of Benson boilers have a serial arrangement of furnace sidewall heating surface, the water flowing upward in all cases to return to the lower header of the next section through downtakes. The mixture leaving the last bundle is composed by volume of 70 per cent steam and 30 per cent water, which is said to be adequate to hold the dissolved solids in solution.

La Mont Boiler

Although this design originated in the United States, the first La Mont boiler was built in Germany in 1930. From that time on several hundred units have been con-

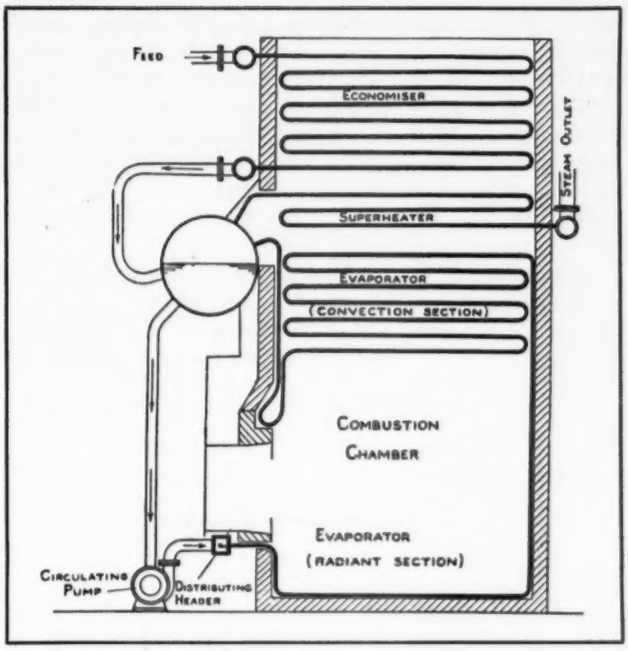

Fig. 3—Circuit of La Mont boiler shown diagrammatically

structed for both high and low pressures. The design lends itself to the highest pressures, although it is also employed for low pressures. Fig. 3 shows the circuit in diagrammatic form, and Fig. 4 a section through a 100,000-lb per hr unit.

This boiler is of the "spill-over" type as reference to the illustration will disclose. The evaporator tubes require the use of orifices to insure equality of distribution in the parallel circuits of the evaporator section. The boiler tubes are $1^1/_2$ in. diameter. It has been found that the nozzles at the tube entrances are essential and that in their absence the circulation through the evaporator tubes would have to be substantially increased to avoid tube damage, with consequent increase in the power requirements.

Because of the small tube units containing many bends, internal cleaning cannot be effected by mechanical methods. Acid solutions containing metal-attack inhibitors must be used. It was originally felt that the inherent high rate of circulation would permit operation with less favorable water than that required by natural-circulation boilers but experience has failed to confirm this. Elimination of solids is effected by blowdown from the drum, similar to natural-circulation boiler practice.

The La Mont boiler contains substantially more water than once-through drumless boilers so that it has the advantage of not requiring such critical control. The pump requires 30 to 40 lb pressure differential which corresponds to about 15 per cent of the auxiliary power or about 0.6 per cent of the boiler output.

The Loeffler Boiler

This design[3] is of particular interest since there is no transmission of heat from flue gases directly to vaporizing water as in the case of other boilers. Instead, the heat is delivered first to a radiant superheater, then to a convection superheater, followed by an economizer and air heater. A part of the steam after passing through the two superheaters serially is delivered as product, the balance (about three to four times as much) is delivered to three evaporator drums and gives up its heat as it bubbles into the drum contents to form saturated steam which is circulated back to the superheaters by a variable-speed pump. The schematic arrangement is shown by Fig. 5. A single smaller drum is provided below the vaporizing drums to serve as a point of accumulation of sludge which is removed by blowing down.

The pump delivers steam to one of the upper front wall headers from which it passes through half of the front wall tubes in a downward direction, then across half of the bottom or grate tubes, then upward through

[3] For a complete description see *Engineering* (London), June 29, 1934.

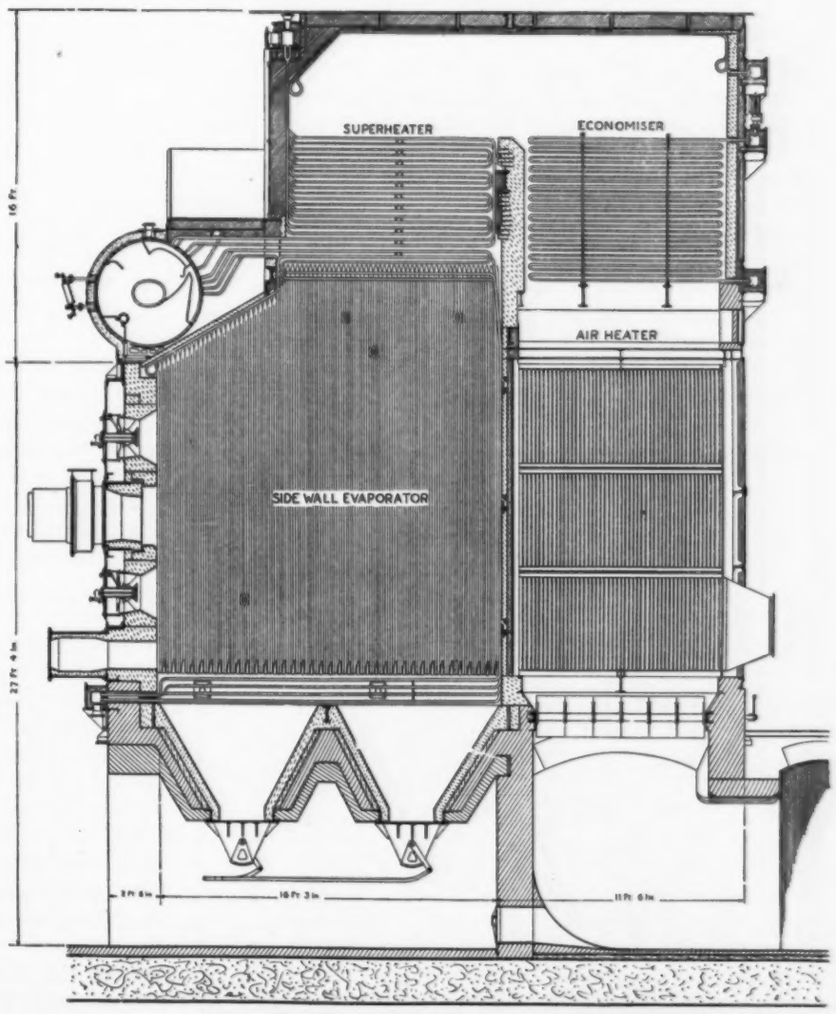

Fig. 4—Section through 100,000-lb per hr La Mont boiler

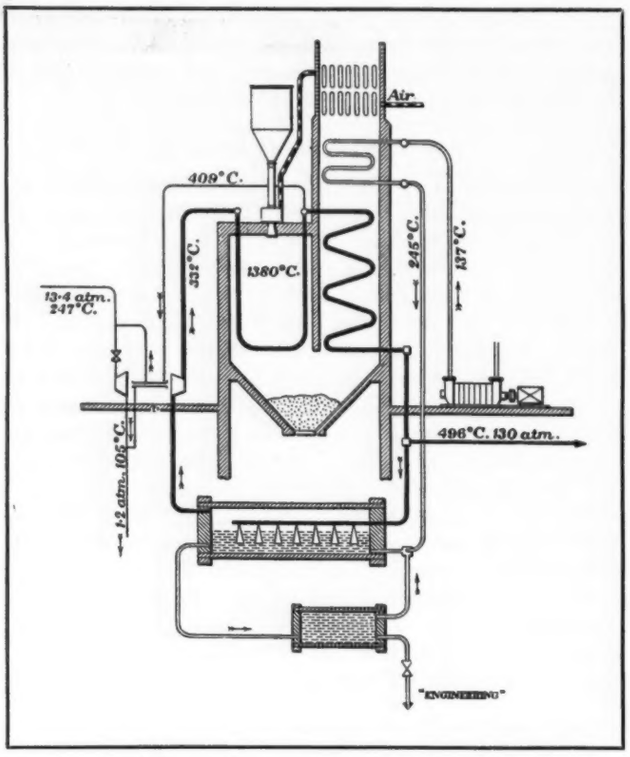

Fig. 5—Schematic arrangement of Loeffler boiler

half of the rear wall tubes. From here the steam is returned to the other front wall header to pass through a similar circuit composed of the other half of front, grate and rear wall tubes. The steam is then caused to flow up and down through the side wall tubes and to the convection superheater. The temperature leaving the radiation superheater is within about 50 to 60 deg F of final temperature. At high loads the temperature difference would be greater.

It is claimed that the boiler is well adapted to poor feedwater and that it has operated without appreciable deposits in the superheaters with drum concentrations as high as 8000 parts per million salt concentration. The steam leaving the circulating pump is slightly superheated, reflecting the work done by the pump. For a 1900-lb pressure unit the pump requires about 4 kw per 1000 lb of steam generated or about 3 per cent of the total output. The pressure drop from pump to delivery, which would represent the loss of pressure through the superheaters, is about 70 lb. Pressure at delivery is about the same or slightly above that in the evaporating drums.

It is apparent that neither the elimination of salts nor suitable control offers any novel problems. The heat storage of the drums is considerable so that regulation need not be close. Control of superheat is effected by increasing or decreasing the pump speed, thus raising or lowering the transfer rate in the convection superheater. The average rate of heat absorption in the radiant superheater is about 40,0000 Btu per hr per sq ft and about 7000 Btu in the convection superheater. Economical application of this design is limited to high pressures, else the increasing steam volume at lower pressures would result in excessive pump power consumption.

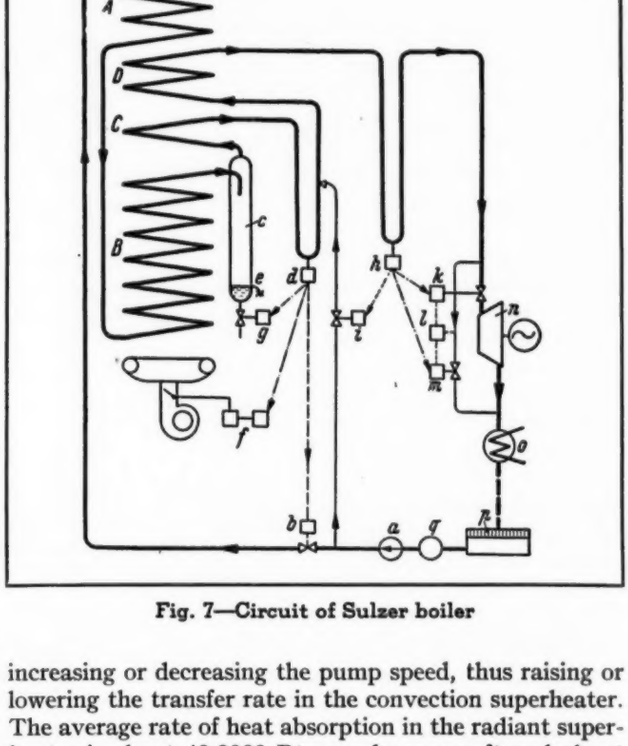

Fig. 7—Circuit of Sulzer boiler

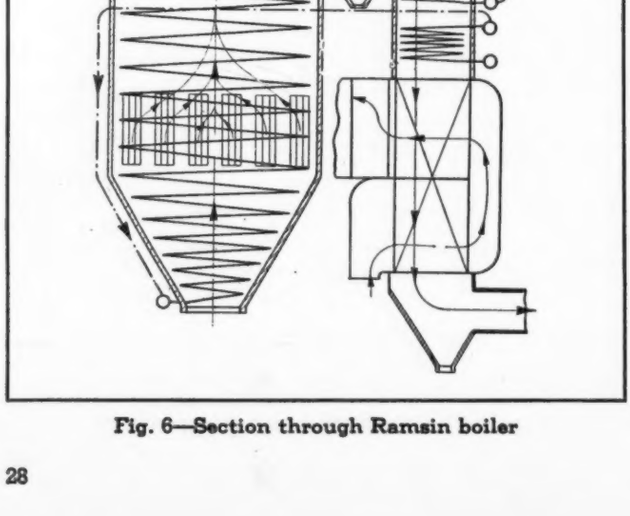

Fig. 6—Section through Ramsin boiler

The Ramsin Boiler

This design[4] is similar to other once-through-flow boilers. After passing downward through the economizer bank (see Fig. 6), the feedwater moves upward through forty spirally arranged parallel circuits, rectangular in section, which line the hopper bottom to form a radiant water heater. In the illustration, the circuits are shown diagrammatically by a single line. The top two-thirds of the coils which extend upward in the combustion chamber form steaming surface, the pitch of the tubes being increased in the upper part of the furnace. Tubes in the lower, or water-heating section, are 1 $^1/_4$ in. diameter and spaced 2 $^1/_2$ in. In the upper, or steaming section, the tubes increase in diameter to 2 $^1/_8$ in. with an increase in pitch to 2.6 in.

[4] See *Die Wärme*, January 15, 1938.

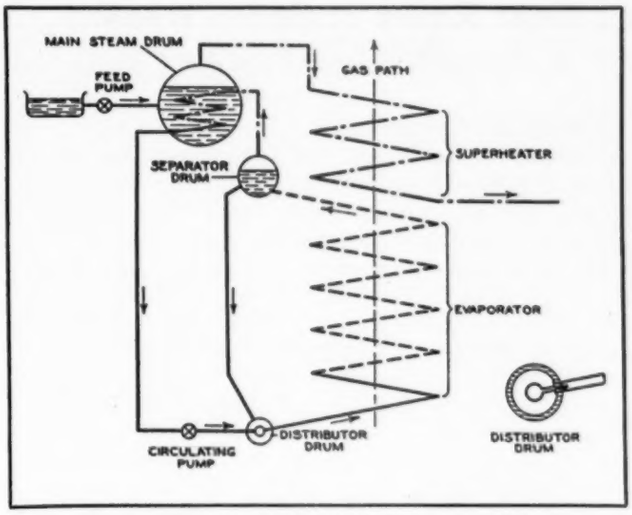

Fig. 8—Circuit of Schmidt-Hartmann boiler

In the convection pass at the top of the unit, the coils change to four horizontal rows of ten tubes each, thence into two rows of twenty tubes each at the entrance to the convection section. These forty tubes then turn back, proceeding to the front of the unit along the sides, then turn up to form the roof and enter the saturated steam header. The steam passes from here through the convection superheater. It would seem that deposits of salt would occur in the upper portion of the furnace tubes or in the convection pass tubes from which its removal by periodic shutdown would be required.

Critical control equipment must be furnished as the water-storage capacity is small. A novel arrangement has been provided for control which consists in establishing a correct proportion between the reciprocating feed pump and the oil fuel pump. When operation is on coal, in combination units, the proper relation is maintained by driving the coal feeder through a liquid motor, the oil being returned to the pump. The arrangement does not, however, improve the need for critical control.

The pressure drop from the economizer inlet to the superheater outlet is considerable, amounting to several hundred pounds. Equality of distribution between the various parts is obtained by the serial arrangement of tubes from various zones. In this manner it is contemplated that any circuit will substantially participate in all of the differences that exist in the combustion chamber and thus will form a resistance equal to that of any other part.

The Sulzer Boiler

The circuit of this design, shown diagrammatically by Fig. 7, comprises a once-through-flow arrangement with continuous blowdown for elimination of the greater part of the salt content. The flow passes from the feed pump through the economizer, *A*, in the usual fashion, thence through the radiant heat absorbing furnace wall tubes *B* and out to a separator *c*. The regulators maintain a surplus of water in the mixture leaving the radiant surface, a part of which is blown down through suitable apparatus for conservation of the heat content of the liquid. The balance of the water and the steam is passed to convection steam surface *C* for complete vaporization of the liquid. From this convection surface the steam is brought out to a thermostat *d* for control purposes as explained more fully below and then taken through the superheater *D* and delivered to the point of use.

The diagram[5] shows the essentials of control. The amount of surface installed at *C* between the separator and the first thermostat, being limited, reflects changes in furnace gas heat content quickly and affords the necessary sensitivity. As the temperature at the control point *d* becomes excessive, three changes are effected: first, the undergrate air supply at *f* is reduced to diminish the rate of combustion; second, the amount of blow-down at *g* is reduced, resulting in an increase in the level of the water in the separator and a greater carryover of moisture; and third, the feedwater regulator *b* is adjusted to allow increased water flow. All of these actions work to restore the temperature.

In addition to the sensitive control described, a second regulating means is provided. This acts upon the basis of the final steam temperature at *h* to allow greater or lesser amounts of water, controlling at point *i*, to be injected into the steam before passing to the superheater. In addition, this thermostat protects the prime mover against excessively high or low temperature by suitable bypass valves as shown at *k* and *m*. The second regulator and the provision for water injection is required to provide constant steam temperature, a requirement which is not accomplished by the first regulator alone.

Schmidt-Hartmann Boiler

This boiler, which is said to be particularly well adapted to poor water, is in effect a double unit. In general, applications have not included any apparatus

[5] Taken from a description by C. Veit in *Zeitschrift* des Vereines Deutscher Ingenieure.

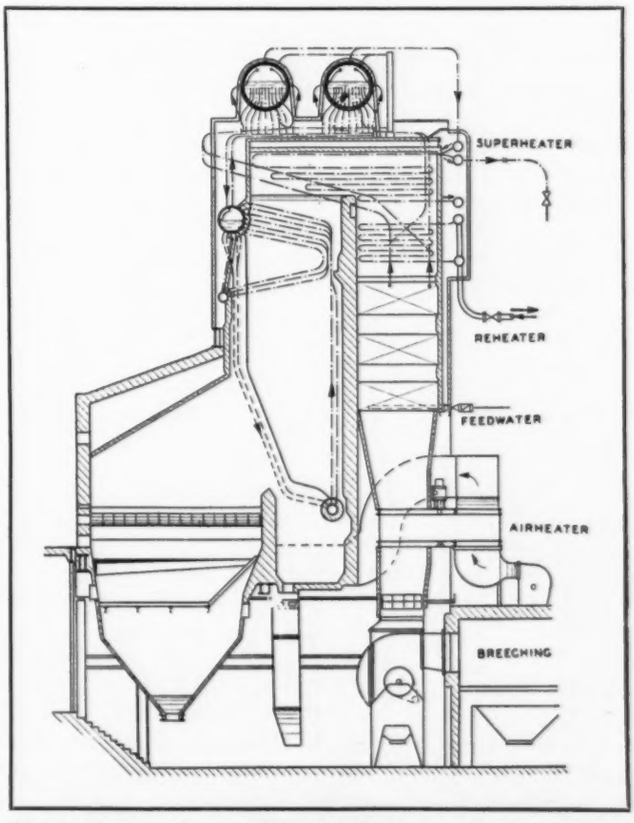

Fig. 9—Section through Schmidt-Hartmann boiler designed for 1500 lb pressure

for assisting circulation, although such an addition could be made if natural circulation should prove inadequate at the higher pressures with high furnace duty.

Fig. 8 shows the circuit in diagram form while Fig. 9 shows the application of this design for a pressure of about 1500 lb per sq in. Water in the furnace walls, convection pass and drum interchanger is kept separate from the feedwater and used repeatedly. Feedwater enters the uppermost drums and is vaporized by the steam produced in the first circuit from distilled water. This steam, at lower pressure than the primary steam, is superheated in the conventional manner. High transfer rates are inherently obtained in heat exchangers between vapor condensing and liquid evaporating. Ample surface is provided so that considerable scale accumulation may take place without interference with regular operation.

Tubes of about $1^3/_4$ in. diameter are used in the primary circuit. This fact is of considerable interest when it is remembered that natural circulation is employed. On the other hand, satisfactory tube size must be considered in the light of furnace tube absorption rate. Information at hand indicates that this is not over a third as much as current American practice. This same feature may explain the survival of the tubes at the top of the combustion chamber which are called upon to receive both radiated and convected heat and, at the same time, are afforded only a small head for circulation. It seems reasonable to suppose that operation at high rates of liberation would require the use of larger boiler tubes as well as a rearrangement of the convection surface at the top of the combustion chamber to afford the necessary increased head.

Conclusions

From technical literature abroad, which is replete with references to forced-circulation boilers and the more fragmentary information which has appeared from time to time in this country, many American engineers are more or less familiar with these designs. It was felt, however, that the foregoing review might serve to interpret the application of these types in the light of conditions existing here and abroad as an explanation of why they have made less headway in this country, without attempting to predict what the future may hold.

Heat Rates of Some Recent Installations

The E. E. I. Prime Movers Committee report on "Combustion" which has just been issued, contains a table of data on a number of new stations and superposed extensions to existing stations. From this the accompanying tabulation is taken.

It will be seen that in most cases the net station heat rates agree closely for similar steam conditions. Slight differences in the case of superposed installations are accounted for by variations in steam conditions and water rates of existing units employed in the topping arrangements. While the performances in some instances are actual, in others they are based on design data, inasmuch as they were furnished before operating data were available.

Boiler efficiencies, as given in the original table, vary from 85.6 per cent in the case of the 12th Street Station, with 410 F gas outlet temperature and a steam pressure of 860 lb, to 88.9 per cent in the case of Waterside Station, with an outlet temperature of 292 F and 1280 lb steam pressure. Although both stations burn similar coal, its delivered cost at Waterside is higher, which differential seems to have justified a larger investment in heat recovery equipment. The difference in gas temperature to the stack also may indicate high base load operation at Waterside against variable load at 12th Street.

While heat-release figures for pulverized-coal-fired boilers of the same steaming capacity convey an approximate comparison of the size of the furnaces and also the intensity of combustion, the Committee points out that such comparison is unsatisfactory since it gives no clue as to performance. It points out, however, that in furnace design the Btu release per cubic foot per hour is a factor of some importance because it is directly associated with and affected by the fundamental relationship of furnace dimensions, rates of water-wall heat absorption, coal characteristics and furnace gas-outlet temperatures.

So far, nothing conclusive has resulted from the operation of these central-station combustion systems to indicate how high rates of heat release may go in the future.

Data on New Stations or Extensions

Station	Capacity Added, Kw	Steam Press. at Superheater Outlet, Lb Gage	Steam Temp., F	Firing	Fur. Heat Release, Btu per Cu Ft	Condensing or Topping	Net Station Heat Rate Btu/Kw/Hr	Remarks
Port Washington	80,000	1350	850	P.C.	16,000	Cond.	10,835 (1937)*	Reheating
Cahokia	75,000	330	740	P.C.	25,200	Cond.	13,920 (design)	For single unit only
Springdale	50,000	1270	935	P.C.	43,800	Top.	11,800 (design)	For superposed section only
Waterside	50,000	1280	900	P.C.	30,200	Top.	11,560 (design)	For superposed section and one unit only
Schuylkill	50,000	1350	910	P.C.	27,300	Top.	11,800 (actual)	For entire plant
Essex	50,000	1350	950	P.C.	27,000	Top.	11,680 (design)	For entire plant
Logan	40,000	1385	925	P.C.	29,800	Top.	12,600 (design)	For superposed section only
West End	35,000	1275	910	P.C.	52,000	Top.	...	...
New Castle (Pa.)	35,000	840	900	P.C.	24,500	Cond.	11,940 (design)	Based on single unit
So. Meadow	35,000	875	910	P.C. or Oil	25,800	Cond.	11,600 (design)	Based on single unit
Cumberland	30,000	850	830	P.C.	25,750	Cond.	12,360	...
Conners Creek	3–30,000 2–60,000	645	840	Stoker	50,400	Cond.	12,440 (1937)	...
Fisk Street	30,000	1275	910	P.C.	31,600	Top.	12,140 (design)	For superposed section only
Millers Ford	25,000	1250	900	P.C.	39,200	Top.	13,850	Entire plant
12th St. (Richmond, Va.)	12,500	860	835	P.C.	29,000	Top.	13,650	For superposed section only
Omaha	10,000	1225	925	Stoker	34,350	Top.	12,900	For entire plant
Elm Street	10,000	800	875	P.C.	22,700	Top.	12,470 (design)	For superposed section only

* The figure for 1938 was 10,788 Btu per kwhr.

Deaerators and Their Applications

By E. S. DAUGHERTY and C. E. JOOS

Cochrane Corporation, Philadelphia

Following a discussion of the chemistry of corrosion and the theory of deaeration, the authors explain the operation and application of different types of deaerators and include typical layouts for central stations and industrial plants wherein such apparatus forms a part of the feedwater cycle.

THE necessity for removing dissolved oxygen and carbon dioxide from boiler feedwater in order to prevent corrosion is now generally recognized. In the corrosion of iron or steel in contact with water, the first step is that the iron goes into solution, forming a definite, soluble compound of ferrous hydrate $Fe(OH)_2$. In the absence of other influences, this action continues until the water becomes saturated with hydrate. The chemical relations are expressed by the following formula:

$$Fe^{++} + 2H_2O \longrightarrow Fe(OH)_2 + H_2^+$$

Dissolved oxygen, if present, at once combines with the ferrous hydrate to form the insoluble ferric hydrate, $Fe(OH)_3$ or rust, according to the following formula:

$$2Fe(OH)_2 + O + H_2O \longrightarrow 2Fe(OH)_3$$

By thus removing the ferrous hydrate from solution, the oxygen prevents saturation from being attained, and iron may, therefore, continue to enter into solution as ferrous hydrate, to again be precipitated as ferric hydrate; these reactions proceeding until either the metal or the dissolved oxygen has been entirely consumed.

Ferrous hydrate is alkaline and the rate at which it will go into solution depends upon the acidity or alkalinity of the water, as expressed by its pH value, which is a function of the ratio of hydrogen ions, H^+, to hydroxyl ions, OH^-, and is a measure of the active acidity or alkalinity and, therefore, of the rate at which corrosion will occur. The pH value, or hydrogen ion concentration, should not be confused with the amount of acid, which determines the total possible corrosion. For example, a dilute solution (5 to 10 per cent) of sulphuric acid, in which there is a high degree of dissociation of hydrogen ions, expressed by a low pH value, attacks iron actively, while concentrated sulphuric acid, in which there is little ionization, is ordinarily shipped in plain steel tanks without damage.

In neutral water the hydrogen and hydroxyl ions are equal in number and at ordinary room temperature the degree of dissociation is expressed numerically by a pH value of 7; in acid waters it is less than 7, and in alkaline water more than 7. Water otherwise perfectly pure, as for example condensate returns, may contain carbon dioxide, CO_2, in solution which, by combining with the hydroxyl, or OH, ions to form carbonic acid, H_2CO_3, leaves an increased proportion of uncombined H ions and a low pH value, as 5.6 to 6.5. Such water, with oxygen also present, is highly corrosive to iron and steel. On the other hand, properly conditioned boiler feed makeup, containing possibly considerable amounts of dissolved salts, but alkaline in reaction and having a pH value of 8.5 to 9.5, is not corrosive if properly deaerated. The pH value of a saturated solution of ferrous hydrate, which by definition would be entirely non-corrosive, is 9.6. It is, therefore, apparent that, for control of corrosion, the steam plant operator is interested both in the removal of carbon dioxide in order to increase the pH value and the elimination of dissolved oxygen. Both results are attained by efficient deaeration, as may be seen by reference to the accompanying table showing the results obtained in testing a large capacity deaerator.

PERFORMANCE OF ATOMIZING DEAERATOR AT VARIOUS LOADS

Water leaving heater 1000 lb/hr	Oxygen cc per liter	Water entering deaerator: Free CO_2 ppm	HCO_3 ppm	CO_3 ppm	pH ppm,
54	7.5	33	29	0	6.0
87	7.4	36	27	0	5.9
108	7.7	32	30	0	6.0
117	7.6	34	31	0	6.0
162	7.3	34	31	0	6.0
189	7.7	20	27	0	6.2
204	7.5	30	27	0	6.2
243	7.6	31	28	0	6.1
273	7.5	32	22	0	6.2
		Water leaving deaerator			
54	Less	0	15	6	8.8
87	than	0	13	6	8.7
108	0.01	0	16	6	8.7
117	cc	0	21	4	8.7
162	per	0	12	8	8.7
189	liter	0	10	4	8.7
204		0	16	6	8.7
243		0	14	6	8.7
273		0	22	4	8.7

As may be seen, the deaerator removes all free CO_2 and some of the half bound CO_2, thereby increasing the pH value appreciably, demonstrating that the deaeration process is beneficial in ways other than mere removal of oxygen.

Theory of Deaeration

The removal of dissolved gases from water is simple in theory. Gases are soluble in water in direct proportion to their respective pressures. In a mixture of gases, or of gases and vapors, the pressure exerted by any one gas is known as its partial pressure. The atmosphere, for example, consists of 21 per cent by volume of oxygen. The average atmospheric pressure at sea level is 14.7 lb

per sq in., and the partial pressure of the oxygen therein is 0.21 × 14.7 = 3.087 lb per sq in. In a mixture of steam and air at 14.7 lb per sq in. and at a temperature of 210 F., the pressure of the steam vapor, as given in the steam tables, is 14.123 lb per sq in., while the remainder, or 0.577 lb, is air pressure. If the air is of normal composition, the partial pressure of the oxygen is then 21 per cent of 0.577, or 0.1212 lb per sq in. The specific solubility of gases under constant partial pressure varies somewhat with the temperature and in Fig. I is shown the solubility in water of oxygen from ordinary air at various total pressures. This chart may also be used to determine the solubility of oxygen from a mixture in which the partial pressure considered is that of the oxygen alone, for which purpose the values should be multiplied by 4.75.

When water is boiling, the pressure of the water vapor at the free surface of the water becomes equal to the total pressure of the atmosphere. No other gases can then have a partial pressure at that point and their solubility is consequently zero. The first step in deaeration, therefore, is to raise the water to the boiling temperature corresponding to the pressures acting upon it. This, however, only renders gases insoluble; it does not insure their removal, since a considerable time is

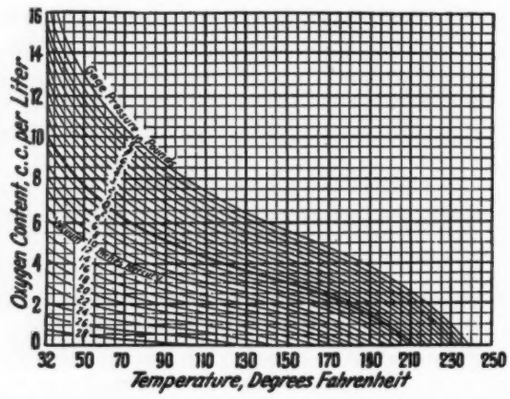

Fig. 1—Solubility of oxygen in water exposed to ordinary atmosphere at given pressures

required for molecules and small bubbles of gas in the interior of the water to reach the surface by diffusion or convection. Moreover, as soon as the water is slightly cooled, or the total pressure slightly increased, the gases will again dissolve in the water. The truth of this statement will be appreciated by anyone who has tried to render water strictly oxygen-free by boiling it in a beaker. In order to obtain water that will test zero by the latest precise methods, prolonged and vigorous ebullition is necessary.

In order to escape from the mass of water, the gases which have been thrown out of solution must travel from the interior of the mass to the surface to avoid the building up of a counter partial pressure at that point. The rate at which complete removal occurs depends upon three principal factors, namely: the concentration gradient, measured by the partial pressure of the gas at a given point in the liquid divided by the distance from that point to the surface, the total area of surface, and, finally, the commotion or turbulence in the surrounding atmosphere which serves to carry the gas away. Obviously, the more the water is broken up into drops or spread out in thin films, the shorter will be the distance to be traversed by the gas molecules within the water and the greater will be the surface presented for their escape, while a blast of steam at the surface will prevent the building up of a partial pressure of the gases.

Principal Kinds of Deaerators

Engineers in attacking the problem of deaeration have availed themselves of three basic methods, namely: reboiling, flowing over trays or pans and atomization. In the reboiler deaerator, oxygen is removed by causing steam to bubble through the water, this steam either being generated by a closed heater element located in the storage space of the deaerator or by blowing the steam into the water through a multitude of small orifices. Usually the water has previously been heated to within 2 to 3 deg F of the steam temperature by means of trays or jets. The possibility of water hammer where steam is forced into large masses of water of lower temperature and the possibility of back flow of water into the steam supply pipes, must both be provided against. For these reasons this type of deaerator has not been widely adopted in the United States. The danger of water hammer in a reboiler deaerator is especially noticeable when starting up the apparatus, as the steam is at the beginning forced into the mass of cold water in the storage space, which must first be brought to the boiling temperature before steam can reach and heat the incoming water in the preheater section. To avoid this difficulty the expedient has been adopted of delivering the steam during the initial starting up period directly to the preheater until the main body of water in the storage space is all hot water, after which the path of steam is changed to blow or bubble through the heated water in the storage space. Meanwhile, however, the boilers will have been supplied with cold, undeaerated water.

The tray type of deaerator is now used very widely because of its simplicity, low cost and effectiveness. In this apparatus the water is spread out over baffles or trays in thin films in such a manner as to provide intimate contact of the steam with the water, through agitation, and a definite path for the steam, so as to remove the gases from the apparatus as rapidly as they are discharged from the water. It differs in these respects from the older and more primitive open heater which, however, was a fairly good deaerator when properly vented.

A number of methods for circulating the steam have been utilized, one of which may be described as counter current or counterflow, the steam being introduced near the bottom of the deaerator and flowing in a generally upward direction contrary to the downwardly flowing water. In another method, known as parallel flow, the steam is introduced into the top of the deaerator and flows downward with the water to be deaerated. In the counterflow deaerator, steam spaces must be liberal in order to avoid appreciable pressure drop caused by the flow of all of the steam through the lower sections of trays, where very little is condensed because the water at this point has already been heated nearly to steam temperature. With such apparatus, it is obvious that the tray spaces must be made amply large, with sufficient openings to prevent undue disturbance of the downflowing water by the upflowing steam.

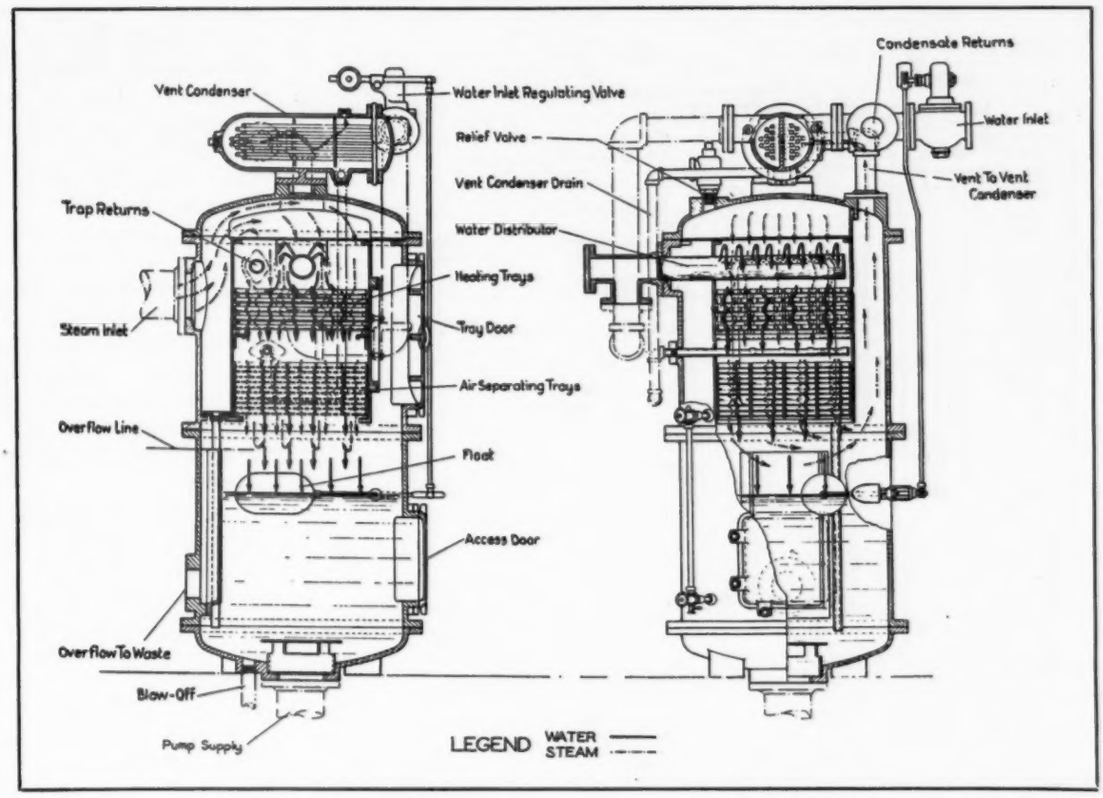

Fig. 2—Downflow deaerator with component parts indicated

These difficulties are overcome in the parallel flow deaerator illustrated in Fig. 2. The water is introduced by spraying it at the top of the heating chamber, at which point the steam is also introduced, the water being heated nearly to the final temperature at this point. Water and steam then travel downward through the stacked trays, the upper trays being openly spaced to accommodate the initially larger volume of steam, while the lower trays are narrow, closely packed and so slotted that they have a great aggregate length of spilling edge. This provides for thin streams, thereby reducing the mean hydraulic depth so that escape of the gases liberated through heating is facilitated. In this way a large total of contact surface is obtained in a compact apparatus, and as the flow of steam is in the same direction as that of the water, the steam actually assists the water flow and promotes uniform distribution. Actual comparative tests on apparatus of this type indicate that considerably greater capacity can be obtained in apparatus of the same physical dimensions with the parallel-flow method than with the counter-flow method.

In another form of downflow apparatus, the water is first heated in a separate upper chamber to a temperature above that corresponding to boiling at the pressure existing in the lower deaerating chamber and is then allowed to escape or "flash," in the latter over trays or pans. Any flashing is detrimental, however, from a thermodynamic point of view, since it implies a drop in temperature and the steam must be supplied to the upper heater chamber at a pressure and temperature higher than that existing in the flash chamber. In other words, the water leaves the apparatus at a temperature lower than that at which the steam is supplied. For good deaeration the amount of steam flashed in and vented from the deaerating chamber should be equal to 2 to 3 per cent of the water deaerated, in order to keep the partial pressure of liberated gases in the vented mixture at a suitably low value. In order to produce deaerated water at 214 F, steam must be supplied to the deaerator at 239 F. If this steam is extracted from the turbine the considerable amount of work that could have been done by expansion in the turbine is sacrificed. Or, if auxiliary exhaust is used this type of deaerator does not use as much steam nor deliver water at as high a heat level as the types which do not require "flashing." This is one of the reasons why the flash deaerator is now obsolete and why the tray type, operating without terminal temperature difference, has taken its place.

Deaeration by Atomization

Another development in the field of deaeration utilizes the principle of atomization, as exemplified by a deaerator designed for marine service. In this apparatus the water is first sprayed into a steam atmosphere, where it is heated to within 2 to 3 deg F of the steam temperature, resulting in the removal of 95 per cent or more of the dissolved oxygen at once. This hot and partially deaerated water is then delivered into a high velocity steam jet, created by the flow of the incoming steam through a spring-loaded nozzle, a velocity exceeding 300 ft per sec being obtained with a pressure drop not exceeding $^1/_2$ lb per sq in. Just as oil is broken up in a steam atomizing oil burner, the water is dispersed into a fine mist, resulting in great total water surface and a minimum distance to be traversed by gas molecules in reaching the surface of the droplets. Inasmuch as the water has previously been heated practically to the temperature of the steam, very little steam is condensed

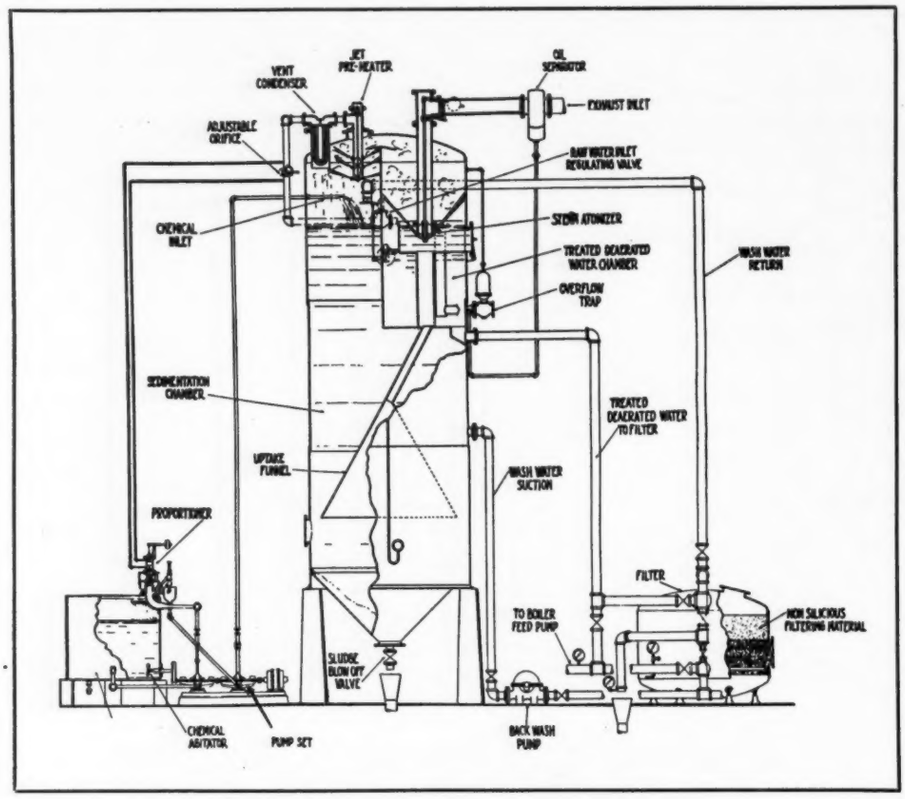

Fig. 3—Arrangement of deaerating softener

in the atomizing process, the bulk of the steam, together with released gases, passing to the spray heater chamber, wherein most of it is condensed. The uncondensed remainder, carrying with it the separated gases, is withdrawn to a vent condenser as in other deaerators.

The atomizing deaerator is particularly well suited for heating and deaerating water containing suspended matter and for deaerating waters which are corrosive because of low pH values, as the few parts coming into contact with undeaerated water are easily constructed of non-corrosible materials. It is also adapted for use on shipboard, where weight and space occupied must be kept at a minimum. It can readily be so designed that the flow of water and the heating and deaerating actions will not be affected by pitching and rolling of the ship, which are highly unfavorable to deaeration in tray apparatus, as they bring about uneven distribution of the water on the trays.

Deaeration by atomization is particularly suitable where deaeration is to be combined with softening by the hot process, as illustrated in Fig. 3. The combination of deaerator and water conditioner results in a more economical and satisfactory arrangement by reason of reduced first cost and better utilization of space and head room. In the combined apparatus the water is first heated by spraying and then, after being mixed with and acted upon by reagents, descends into the sedimentation tank from which the settled water rises through a conical collector to the atomizing chamber, wherein the water overflows onto the steam jet of the atomizer. After this final deaeration the water passes to a filter and then to the boiler feed pumps. Tray deaerators are unsuitable in connection with softeners as the trays would become clogged by sludge or scale.

All types of deaerators, except flash deaerators, are like the open or direct contact heater, in that they heat the water fully to steam temperature, and thus eliminate terminal temperature difference and the loss of thermodynamic availability which it implies. Advantage has been taken of this fact in certain central stations by employing a direct contact heater as one of the stages of a regenerative feed-heating system to raise the overall efficiency of the plant cycle.

Applications of Deaerators

In the application of deaerators, it is to be borne in mind that they accomplish the following two desirable objectives:

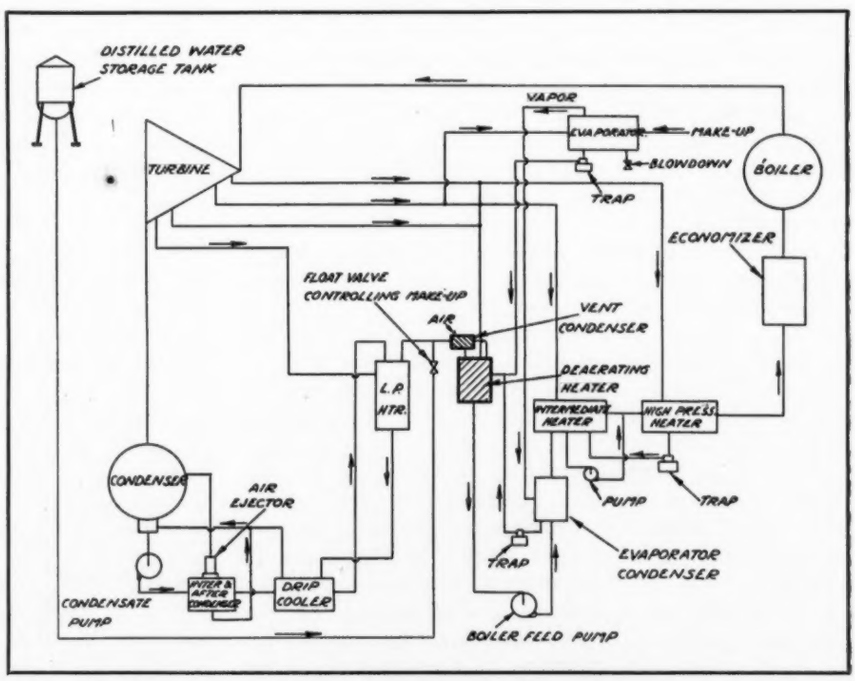

Fig. 4—Feedwater cycle in central station having four-point regenerative heating with deaerator as one stage heater

1. Removal of dissolved gases, to prevent corrosion.
2. Heat the boiler feedwater to the temperature of the steam, thereby gaining a thermodynamic advantage as compared with tubular heaters.

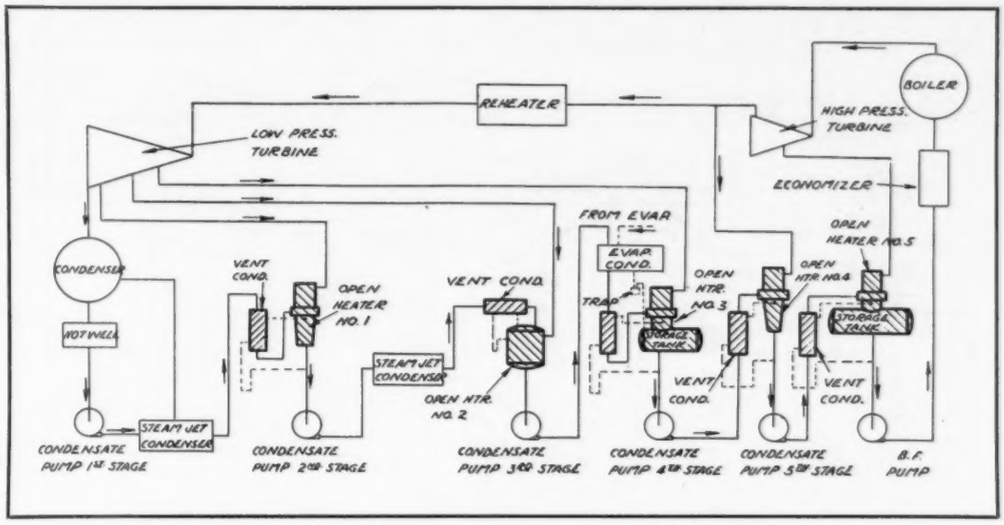

Fig. 6—Heat cycle of a 1200-lb central station with five-stage bleeding

Fig. 4 illustrates the feedwater cycle of a central station with regenerative feed heating from four bleed points, in which the deaerator serves as one of the stage heaters, accomplishing the double objective of removing corrosive gases and of heating the feedwater with maximum efficiency. Fig. 5 illustrates a typical installation in an industrial power plant, in which the deaerator combines the function of heater, gas remover and receiver and storage chamber for condensate returns. Installations of this kind can provide sufficient surge tank capacity so that, in many instances, separate tanks are not required.

Fig. 6 illustrates the heat cycle of a 1200-lb central station bleeding steam from five stages of the main turbine to supply five direct-contact heaters, one of which is a deaerator for removal of dissolved gases. The bleed pressures range from a vacuum on heater No. 1 to 800 lb on heater No. 5. A number of such plants are now in operation and are giving very satisfactory results.

In Fig. 7 is shown the flow diagram of a modern central power plant designed to supply the needs of an industrial plant. Steam is generated at around 1500 lb for producing power, and is bled from the turbine to an evaporator for supplying process steam, which is not returned to the cycle. Steam for heating the water fed to the evaporator is taken from the power cycle and, in order to keep the size of the evaporator to a minimum, as much as possible of the bled steam is returned to the power cycle. To insure that little of the steam used for deaerating the water supplied to the evaporator will be condensed in the deaerator, the water is first heated to within 2 to 3 deg of the steam temperature in closed heater No. 7 and the pressure of the steam supplied to the deaerator is reduced so that its saturation temperature will not exceed that of the incoming water. By this arrangement steam is caused to flow through the deaerator for venting purposes without condensation, thus obtaining all of the benefits of deaeration without consumption of steam. The steam vented from the deaerator to the vent condenser returns to the power cycle.

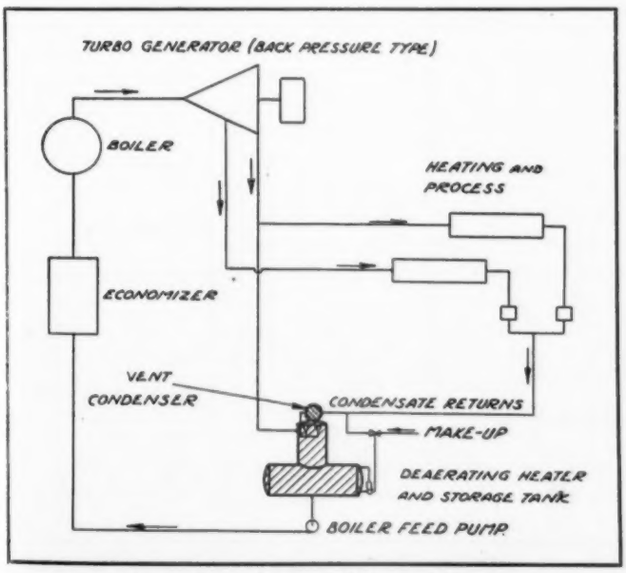

Fig. 5—Typical industrial plant layout

Deaerating Service Water

Where city water supplies are corrosive, hot water service lines of steel corrode rapidly. Such corrosion can be avoided by the use of deaerators which rid the water of corrosive gases while heating it to the temperature of use. In such apparatus it is not desirable for the heating steam to come into direct contact with the water heated because of the possibility that the water may be contaminated by odors or impurities carried by the steam. The steam which performs the actual deaeration is, therefore, generated from the deaerated service water itself by tubular heaters placed in the water-storage space below the deaerating trays.

Such apparatus can be applied effectively in large buildings and Fig. 8 illustrates a deaerator located in the basement, with a sealed storage tank on the roof. In many large buildings the service water is "zoned" in order to avoid excessive pressure at the faucets, in which case the deaerating heater can be placed in the basement and sealed hot deaerated storage provided at the zone levels. In a community central heating plant, where it is advantageous to return the condensate, protection of the condensate return lines against corrosion by condensate of low pH value and generally contaminated with oxygen by infiltration, particularly in heating systems of the vacuum type, is desirable. In such a layout it is necessary to deaerate the water before returning it to the central power plant and it has been

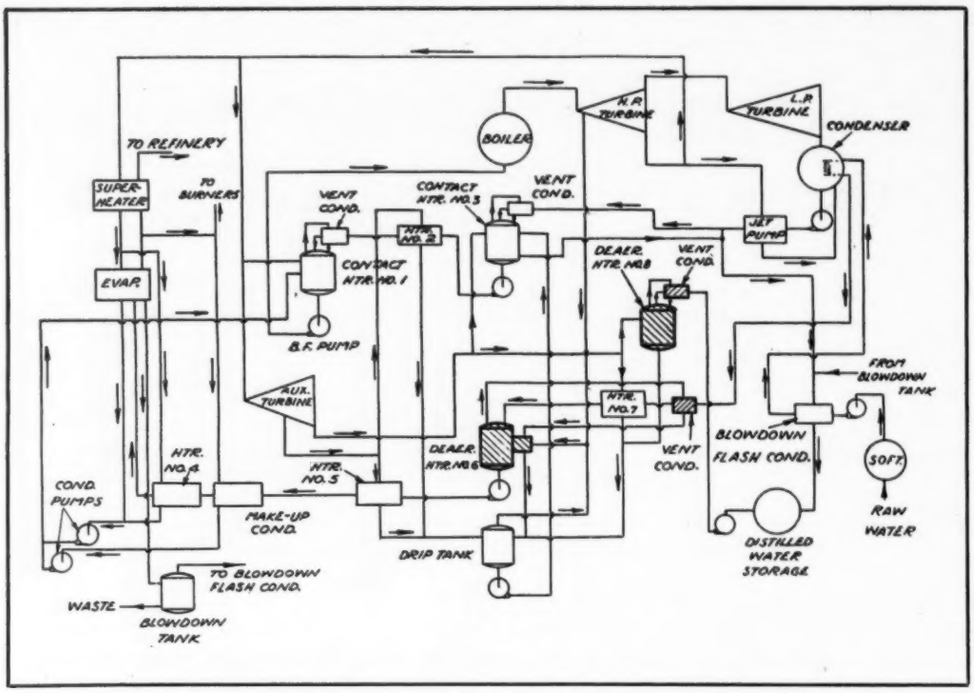

Fig. 7—Diagram of high-pressure plant supplying power and process steam

customary to install independent deaerators with storage tanks so that the deaerated condensate can be pumped or returned to the power plant without fear of corrosion.

Deaeration of Cold Water

Frequently, water which is corrosive does not require to be heated, as where corrosive water is used as cooling water in iron or steel heat exchangers or where corrosive water is to be delivered miles from the source through long pipe lines. In such circumstances satisfactory deaeration can be accomplished by spraying and passing the water over trays under vacuum. Spraying as a means of deaeration was applied with success over 30 years ago for the protection of the Coolgardie (Australia) 30 in., 350 mile long steel pipe line. The water was sprayed into the top, and removed from the bottom of a high, air-tight cylinder in which a fairly high vacuum was established by means of a vacuum pump drawing from the top of the cylinder. More recently the same method has been used by Messrs. Powell and Burns at the Grande Ecaille mines to which water is transported from the Mississippi River 9 miles away through a welded steel pipe. In their apparatus slats were introduced into the cylinder to retard the fall of the water, thereby obtaining a larger total water surface and a longer exposure of the water without the necessity of a very high tank. Carrying a $28^1/_2$-in. vacuum the oxygen content is reduced to around 0.33 part per million. While greater removal could be obtained by carrying a higher vacuum, much larger and more expensive apparatus would be necessary. It was found experimentally that the cost and power requirements of the vacuum apparatus could be kept down by carrying the process out in two stages, but practically it was preferred to remove the remaining oxygen by treating with sodium sulphite.

Vacuum deaerators have been used for deaerating water at temperatures lower than 212 F where steel-tube economizers were employed in power plants and where the water was to be used for service requirements, as in many industrial plants, or in connection with air-conditioning systems.

Chemical Deaeration

Certain chemicals have been used for fixing small quantities of oxygen. Ferrous hydrate has been applied at the Lakeside Station of the Milwaukee Electric Light Company and sodium sulphite is quite generally used to supplement mechanical deaeration. Properly designed deaerators, however, will deliver water of zero oxygen content as determined by the Winkler test and supplementary treatment is not then necessary. To deaerate chemically, water containing relatively high concentrations of oxygen, as 2 to 3 cc per liter or higher, would not be economical, as the cost of chemicals would be prohibitive as compared with the first cost and cost of operation of a mechanical deaerator.

Forms of deaerating apparatus modified to suit special local conditions are widely used and because of their effectiveness, long life and low maintenance and operating costs should form an integral part of almost every feedwater cycle, as they serve the triple purpose of preventing corrosion, bringing about an improved thermodynamic cycle and providing surge capacity for storing hot deaerated water.

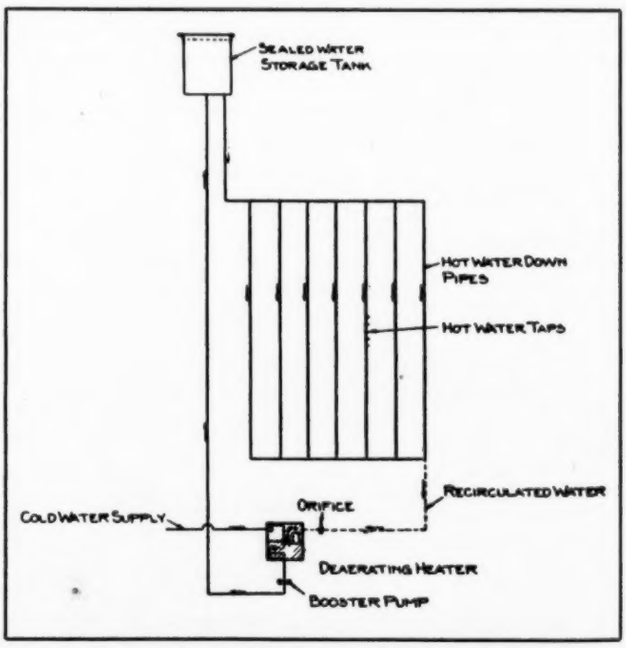

Fig. 8—Deaerator in building basement with pump to supply storage tank on roof

Efficiency Comparisons of Turbine-Generators and Generating Stations

By JAMES F. FIELD

In this paper, recently presented before the Institution of Civil Engineers (Great Britain), the author points out the inadequacy of the simple Rankine cycle as a basis of comparing performance of turbine-generators in which steam is extracted for feedwater heating. He suggests a modification of the Rankine cycle to allow for both reheating of the steam and feed heating by extraction in an assumed infinite number of reversible stages.

IN RECENT years the possibility of increasing the thermal efficiency of the multi-stage turbine by feed heating with bled steam has been appreciated, and the principle of inter-stage steam reheating has simultaneously been introduced in many cases, chiefly to overcome the practical difficulties of expanding excessively wet steam efficiently in the low-pressure stages. The simple Rankine cycle is not an appropriate criterion for this type of engine.

It was recognized by the Institution that such a plant should be comparable with a modified Rankine cycle, in which the ideas of steam reheating and feed heating are introduced to a corresponding degree. The report of the Heat Engine Trials Committee of the Institution in 1927 described modified Rankine cycles, where both ideas were incorporated in an ideal engine in which the whole of the steam flow is reheated at a chosen intermediate pressure, and in which the whole of the steam required for feed heating is extracted at a pressure corresponding to the specified final feedwater temperature. The 1927 report indicated the method of calculating the efficiency of this modified Rankine cycle and of using it as a basis of performance for feed heating and reheating.

The ideal Rankine cycle, modified by a single stage of feed heating, has, however, the disadvantage of not being completely reversible within itself, hence the calculated improvement in thermal efficiency over the simple Rankine cycle is less than is obtained in practice. The discrepancy lies in the idea of bleeding the steam in a single stage rather than in a multitude of stages, so chosen that there is an infinitely small difference in temperature between the bled steam and the feedwater at any particular point; that is to say, that process is reversible. In practice, efforts are made to approach the condition of reversibility by having the turbine bled in several stages, as many as five stages being fairly common.

It is now suggested that the Rankine ideal engine should be modified to allow both for steam reheating and for feed heating by extraction of steam in an infinite number of reversible stages, and that this should be used as the modern ideal standard. Whereas the original Rankine engine consists of a boiler, superheater, cylinder and piston, condenser, and feed pump, the modification consists in adding a re-superheater through which the drop in steam pressure is small, and an infinite number of reversible feed-heating stages.

Thermodynamic Considerations

The efficiency of the original ideal Rankine engine is almost exactly equal to

$$\frac{\text{adiabatic heat drop to condenser}}{\text{total heat received per pound of steam}}.$$

The error in this assumption, which neglects the work spent in the feed pump, is very small and can be ignored except for very high steam pressures. With dry saturated steam the cycle follows the first isothermal reversible operation of the Carnot cycle, and likewise the second adiabatic reversible operation, but in the third operation it proceeds to complete condensation of the steam. In the fourth the water is compressed in a pump to boiler pressure and external heat is added to bring it up to evaporation temperature, in place of the partial condensation and adiabatic compression of the mixture of steam and water up to water at boiler pressure and temperature, as required for the third and fourth operations in the Carnot cycle.

By feed heating up to evaporation temperature it is possible to avoid the taking in of heat by the working fluid below this temperature. With saturated steam the cycle is in effect a Carnot cycle, since the process of feed heating is assumed to be reversible, and the two engines work between the same limits of temperature. Fig. 1 is a temperature–entropy diagram for this process. Assuming that each pound of the working substance is heated from A to B by steam bled from the preceding pound, the work done per pound will be area $ABCDJIA$ minus the area $JCDJ$, or area $ABCJA$, and the heat rejected externally will be area $AJFHA$. All the external heat is added at the upper temperature corresponding to B. The efficiency will then be $\frac{T_2 - T_1}{T_2}$, as in the Carnot cycle. In practice the greater part of the heat is added at the saturation temperature corresponding to the boiler pressure, but a limited amount of superheat can be added, and this heat, being taken in at a higher temperature, is more efficiently used.

In the application of feed heating the boiler is required to furnish a fixed quantity of heat per pound of steam

under certain conditions. For maximum efficiency the aim is the maximum conversion into mechanical energy of the sum of the heat drops available per pound of the working substance. There is, for instance, the expansion of the main body of steam to condenser pressure, and of the several heater quantities to intermediate pressures. Optimum use is made of the limiting steam conditions by arranging that heat is either rejected externally or transferred to the feedwater with the minimum difference in temperature; that is, each part of the cycle approaches reversibility for maximum efficiency.

Steam is bled at a few intermediate points in the expansion, but at higher temperatures and pressures (and at a higher total heat) than it theoretically need have to transmit heat to the feedwater. The surface-type heater requires an economic minimum temperature difference of the order of 10 deg F to transmit the heat through the resistant films at each side of the heater-tube material. Expansion in the turbine is not strictly adiabatic, due to friction and shock, and reheat suffered by the steam due to this effect, being irreversible (Fig. 2*d*), reduces the efficiency of the process. A lesser proportion of the stop-valve steam is bled off, and a greater amount of heat is rejected to the condenser.

It is not convenient with steam bled in a superheated condition to heat feedwater beyond the saturation temperature corresponding to the pressure of the bleed point. An examination of modern steam conditions will show that the highest bleed point theoretically takes place—and in practice certainly would take place (Figs. 2*b*, *c* and *d*)—in a region where the expanding steam is still in a superheated condition. The effect on the diagram efficiency is shown by the line *DB* which bends to the line *BA* after passing the dry saturated region of expansion. Such an engine is not therefore reversible in the method of bleeding superheated steam. Complete reversibility can only be achieved if the steam is bled at the temperature corresponding to the feed temperature (line *ABC*, Fig. 2*a*), and the ideal engine would therefore require additional elements to compress the superheated bled steam isothermally to the corresponding saturation temperature and pressure before actual mixing with the feedwater. It is convenient to imagine the existence of these where necessary, so that the formula for calculating the ideal efficiency is simplified. Fig. 2*b* shows the effect on the diagram efficiency of bleeding all the steam in one stage, and Fig. 2*c* the improvement with five bleed points.

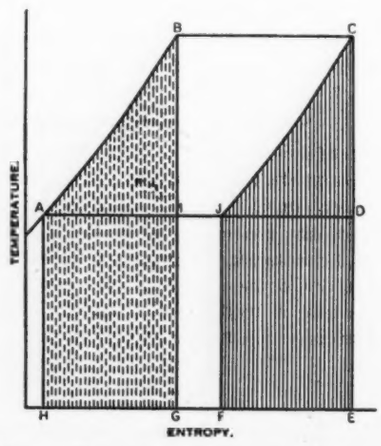

Fig. 1—Temperature-entropy diagram

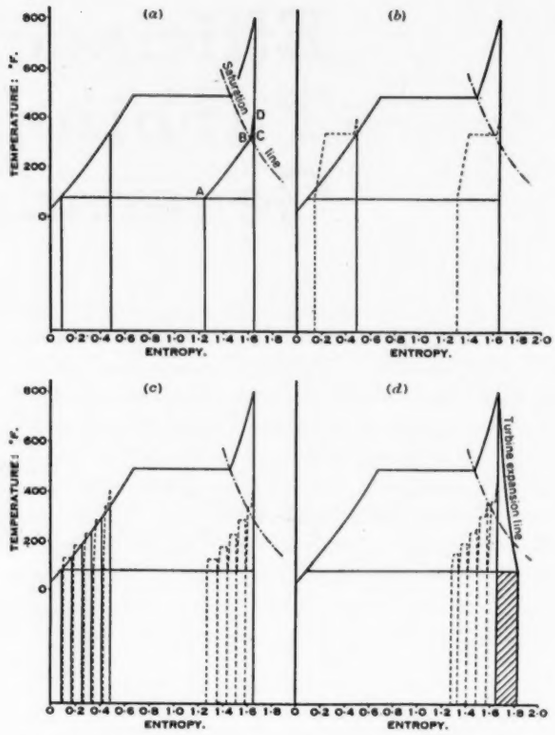

Fig. 2—Comparison of actual with ideal conditions

In the application of reheating it is advisable to bleed steam at the turbine outlet to the reheater, rather than after reheating, since the former process is more conveniently reversible, and only a proportion of the steam passing through the turbine throttle also passes through the reheater. This does not affect the validity of the following formula which assumes the whole of the throttle steam to pass through the reheater (since the ideal engine is completely reversible within itself), and steam can be bled either before or after the reheat point provided it is subjected to reversible compression or expansion if necessary before its heat is added to that of the feedwater. Alternatively, an ideal cycle can be assumed in which no bleeding takes place, but is completed with an adiabatic, an isothermal and an adiabatic, as in the Carnot cycle. This will have the same efficiency as any other reversible cycle working within the same limits of temperature.

Standard of Efficiency

The efficiency of the ideal engine =

$$\frac{\text{(Heat supplied to engine)} - \text{(heat rejected to condenser)}}{\text{(Heat supplied to engine)}}$$

For the modified Rankine cycle, using superheated steam and feed heating, Fig. 3 shows the formula to be as follows:

$$\text{The ideal feed-heating efficiency} = \frac{\text{(Heat supplied under feed-heating conditions per pound of water)} - \text{(absolute temperature of condensate)} \times \text{(difference in entropy between initial steam and feedwater)}}{\text{(Heat supplied under feed-heating conditions per pound of water)}}$$

It will be observed that when the feedwater temperature coincides with the condensate temperature the cycle efficiency becomes that of the Rankine cycle.

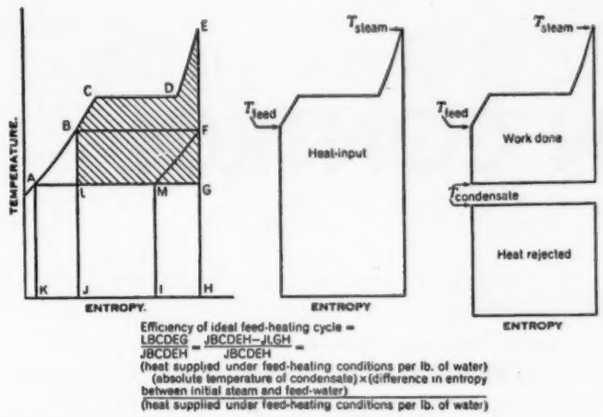
TEMPERATURE.
ENTROPY.
ENTROPY
ENTROPY
Heat-input
Work done
Heat rejected
Efficiency of ideal feed-heating cycle =
LBCDEG / JBCDEH = JBCDEH−JLGH / JBCDEH =
(heat supplied under feed-heating conditions per lb. of water)
(absolute temperature of condensate) × (difference in entropy between initial steam and feed-water)
(heat supplied under feed-heating conditions per lb. of water)

Fig. 3—Diagram for modified Rankine cycle with superheated steam and feed heating

Fig. 4 illustrates the case of intermediate steam reheating, and from the diagram the formula is as follows:

$$\text{Ideal reheating feed-heating efficiency} = \frac{\text{(Heat supplied under feed heating + reheating conditions per pound of water) − (absolute temperature of condensate) × (difference in entropy between reheated, or final reheated,[1] steam and feedwater)}}{\text{(Heat supplied under feed heating + reheating conditions per pound of water)}}$$

[1] If more than one stage of reheating is adopted.

Condensing heat rate = 11041.15 Btu per kwhr
Feed-heating heat rate = 10013.91 Btu per kwhr
Gain due to feed heating = 10.26 per cent

This compares with the ideal-cycle gain of 12.54 per cent.

Application of Formulas to Turbine-Generators and Power Stations

The foregoing formulas are convenient criteria either for the turbine-generator unit complete with feed heaters, extraction, and boiler-feed pumps, bounded by the limits of the stop valve, exhaust-flange, and high-pressure heater outlet, or, may include the boiler and auxiliaries; that is, the station overall performance.

The formulas have been applied to a 70,000-kw turbine-generator operating between 570 lb per sq in. gage, 800 F and 29 in. vacuum, with feed at 340 F. The guaranteed heat rate is 10,014 Btu per kwhr (excluding energy to the boiler feed pump[2]). The ideal efficiency by the formula is 43.62 per cent, equivalent to approximately 7820 Btu per kwhr, so that the efficiency ratio for the steam turbine-generator only, including feed heaters, is 78.1 per cent.

[2] The author considers that since the boiler feed pump is an essential part of the ideal engine of comparison, it should be included in the overall measurement of a turbine-generator.

PERFORMANCES OF POWER STATIONS COMPARED

Station	Date	Steam-pressure: lb per sq. in. gage	Total temperature, F	Vacuum, inches mercury	Feedwater temperature, F	Ideal-cycle efficiency, per cent	Actual overall efficiency, per cent	Ratio of efficiency, per cent
Dalmarnock	1922	250	650	29	150	36.5	16.6	45.5
Barton	1926	350	700	29	220	39.1	20.7	53.0
Deptford West	1930	350	780	29	270	40.4	23.0	56.9
Clarence Dock	1933	400	700	29	270	40.5	24.8	61.2
Dunston "B"	1936	600	800*	29	350	44.7	26.87	60.1
Battersea L. P.	1936	600	850	29	350	44.3	27.63	62.4
Barking "B"	1936	600	800	29	350	44.0	26.54	60.3
Port Washington, U. S. A.	1937	1230	825†	29	427	48.9	31.49	64.5
Brimsdown "A"	1939	1900	930**	28.75	375	49.0	?	?
Battersea "B"	1940	1350	950	29	400	49.3	?	?

* Reheating at 115 lb per sq in. gage to 800 F.
† Reheating at 425 lb per sq in. gage to 825 F.
** Reheating at 175 lb per sq in. absolute to 800 F.
Steam conditions are assumed to be those which obtain at the maximum continuous rating of the turbine.

Application of the Formulas to Feed-Heating Calculations

Many engineers like to think of the adoption of steam reheating or feed reheating as giving a thermal efficiency gain of so much per cent on the equivalent Rankine cycle. This is probably due to familiarity with the easily calculated Rankine cycle. There is also the practical consideration that it becomes increasingly difficult and costly to maintain a specific standard of boiler efficiency as the feed temperature to the boiler rises beyond certain limits, and in high steam pressure developments a compromise in feed temperature has been generally necessary.

Below is given an efficiency comparison for a feed-heating cycle and the simple Rankine cycle, and for turbines of modern designs working under the corresponding steam-conditions. Assumed steam conditions are:

Initial pressure 570 lb per sq in. gage and initial temperature 800 F, exhausting to 29 in. vacuum; feed heating at maximum continuous rating to 340 F.

Using the formula the ideal-cycle efficiency = 43.62 per cent, and the corresponding simple Rankine-cycle efficiency = 38.76 per cent; that is, an improvement is shown on the Rankine cycle of 12.54 per cent.

A turbine-generator built for these operating conditions, with five stages of feed heating, has the following performance:

The formulas have also been applied to the performance of power stations, and Johnstone Wright, Chief Engineer to the Central Electricity Board, at a recent meeting of the Institution compared in diagrammatic form the performance of various power stations as recorded in the Electricity Commissioners' returns and in other published data over the last 20 years. With his permission the author is repeating these figures in tabular form, with a note of the steam-conditions, etc., derived also from published data.

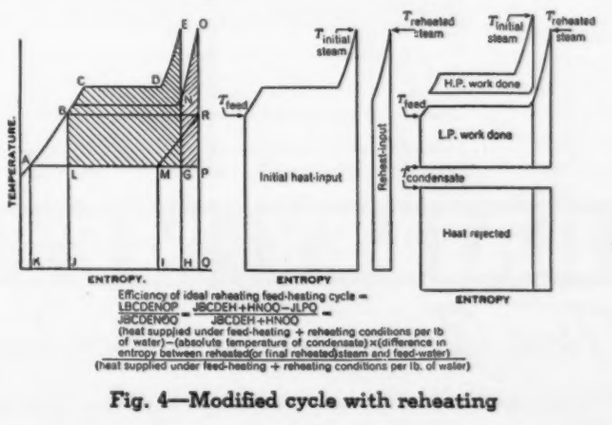
TEMPERATURE.
ENTROPY.
ENTROPY
ENTROPY
Initial heat-input
Reheat-input
H.P. work done
L.P. work done
Heat rejected
Efficiency of ideal reheating feed-heating cycle =
LBCDENOP / JBCDENOQ = JBCDEH+HNOO−JLPQ / JBCDEH+HNOO
(heat supplied under feed-heating + reheating conditions per lb of water)−(absolute temperature of condensate)×(difference in entropy between reheated(or final reheated)steam and feed-water)
(heat supplied under feed-heating + reheating conditions per lb. of water)

Fig. 4—Modified cycle with reheating

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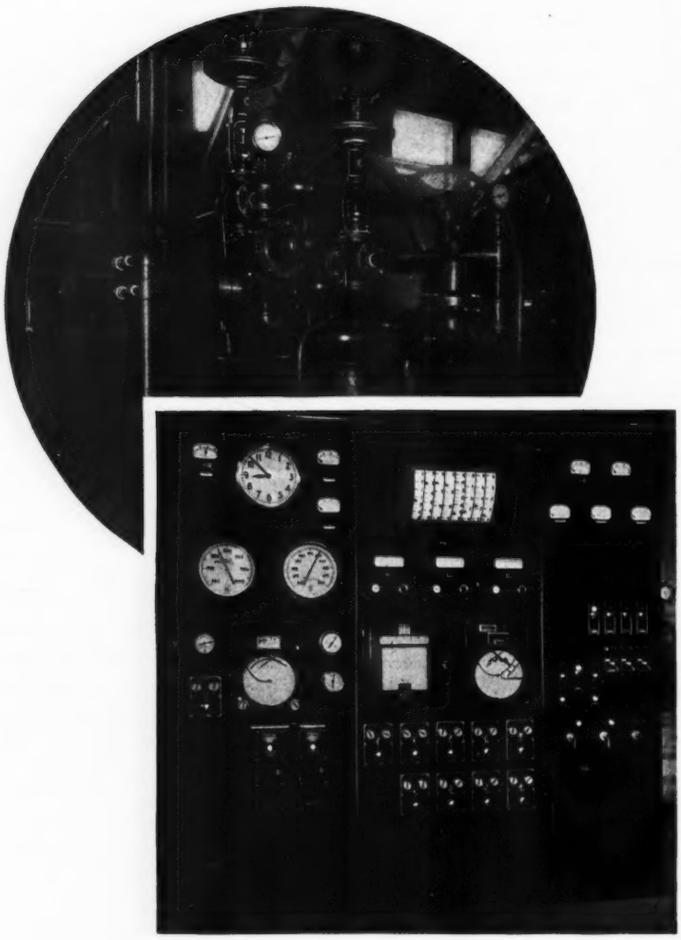

Bailey Boiler Control Panel at Riverside Station showing Master steam pressure recorder-controller and the Steam Flow—Air Flow Bailey Boiler Meter which automatically readjusts the air supply to maintain best combustion conditions at all times.

Bailey Control Valves regulating induced draft fan speed by a hydraulic coupling installed between the fan and the motor.

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Cylinder Oil in Condensed Steam

By A. F. BREWER

Technical Division
The Texas Company

The presence of oil in the exhaust steam from engines, pumps and compressors is detrimental to its direct use for certain processes and for boiler feed; hence, certain precautions are necessary in order to keep this to a minimum. The oil content in the exhaust can, to some extent, be controlled by careful selection and proper application, which features are discussed by the author, as well as the subject of compounding cylinder oils.

THE handling of steam cylinder oils where condensed steam is to be used as a medium for heat exchange is a matter that must be approached with due consideration of the effect which oil in the condensate may have upon the materials being handled or upon the rate of heat transfer. The problem will involve initial prevention of carry-over of such oils as much as possible, and their subsequent removal by passing the condensate through oil separators or grease extractors. It is best to concentrate on studying ways and means to prevent carry-over, with reservation of subsequent separation as an insurance factor.

Just why steam cylinder oils will become so entrained with steam is of decided interest to the power plant operator. Primarily, he must realize that he is usually dealing with emulsifiable products. In other words, oils such as those prepared for average steam conditions will contain a certain amount of animal compound, as degras, tallow or lard oil. This component is necessary to aid in the formation of a lathery lubricating film that will resist the washing action of any moisture contained in the live steam or developed during its expansion by cylinder condensation. That is, the film tenacity is increased by compounding and greater adhesiveness for metals is assured. The more compound used (within due limits), the more adhesive will this lubricating emulsion become. Later, however, it will be more difficult to break and separate from the condensate. Furthermore, the more completely the oil is atomized in its delivery, the greater will be its dispersion; this, however, renders subsequent separation more difficult.

Therefore, the amount of compound is accurately controlled in the manufacture of a steam cylinder oil. Rarely will it exceed ten per cent except where the oil is prepared for use with very wet steam, and normally it will be about half this figure. Present-day practice endeavors to reduce the amount of compound and improve its quality. It is fully realized that the excess use of such material, beyond the amount required to form the necessary emulsion, does not improve the lubricating ability of the oil; furthermore, it may become a detriment wherever the exhaust is used for heat exchange.

The petroleum chemist, in his studies of compounding as an adjunct to steam-cylinder lubrication, is continually in contact with operating performance through reports from the field, and is able thereby to estimate the lubricating value of the various materials available. As a result, it has become practicable to some extent to select cylinder oils of characteristics conducive to subsequent separation from exhaust steam, but which will emulsify sufficiently to assure of dependable cylinder lubrication. This requires, however, the use of selected mineral oil stocks of filtered nature along with high-grade fats. The final viscosity should not be too high, and in application they should be handled sparingly so as to just develop adequate lubrication.

The use of relatively soft boiler water is also an insurance against difficulty. Where the concentration of carbonates or sulfates of calcium or magnesium is high, there will be a tendency of these salts to react with some cylinder oil compounds to form insoluble soaps. If this develops during priming or foaming, deposits on the tubes will be accelerated.

Guarding against Contamination

The extent to which oily or fatty matter may be objectionable in steam or condensate used for feedwater or process heating will, of course, depend upon the extent to which direct heating is practiced. This is understood, normally, to involve open feedwater heaters, and dyehouse or rinsing vats in the textile industry, for example, where the steam may be led directly into the water or solution to raise its temperature. It is usually the cheapest sort of installation to make and often preferred for economic reasons.

Oil or grease spots are detrimental to any textile material whether in skein or piece form. When we approach the finished product, as for example in piece-dying, all the more care is necessary to prevent oil contamination, for fewer finishing processes are involved with many types of goods. The lubricating engineer guards against this in recommending lubricants for mill machinery; the power plant must, in turn, cooperate in preventing direct contamination through contact with oily matter in the exhaust steam. Hence, the advisability of using water-softening equipment wherever markedly hard water only is available, and the use of steam-heating coils in bleaching and rinsing vats, or oil separators wherever direct heating is practiced. These latter means are highly efficient in their ability to separate oil from steam. They must, however, be chosen to conform to the operating conditions, the probable maximum amount of oil which the exhaust may contain, and be cleaned regularly in service.

There are, of course, some heat-exchange operations where the exhaust or condensate does not come in direct contact with the materials. Here, we must be more concerned with deposition of fatty matter on the surfaces of the walls or tubes of the exchanger in order to reduce the rate of heat transfer. Nevertheless, a very thin layer will so insulate the surface that not only will there be a high loss in the rate of heat exchange, but also the rise in temperature of the metal may be so excessive as to cause blistering or burning and subsequent failure.

Incidentally, indirect heating, by means of exhaust steam can be brought about in the paper industry in the operation of certain types of calender stacks. This step in the manufacture of paper is more or less an ironing process, the purpose being to develop a finish to the surface of the paper after it has left the dryer end of the paper machine. The calender stack rolls which perform this function are steam heated, and if too high temperatures are not required, exhaust steam of a few pounds pressure can be used, this being circulated through the rolls via steam joints.

Selection of Steam Cylinder Oils

With an understanding of the determinents which may result from oil in exhaust steam, it will now be advisable to consider the manner in which cylinder oils should be selected to assure of minimum carry-over. Cylinder lubrication will involve reciprocating steam engines, pumps and compressors. The nature of the steam and type of lubricating equipment have the most influence upon the choice of any such oil. In other words, this must be based upon the pressure, temperature, quality (or moisture content), the velocity, whether feedwater treatment is employed, the possibility of priming or foaming, the type of atomizer, means available for oil delivery and the distance of the latter from the throttle. All these are mechanical in nature. From a more or less chemical viewpoint, one must consider the readiness with which the oil will atomize (based on its degree of refinement), the probable amount of emulsification required, and the rate at which separation of oil from the exhaust will occur.

To meet any combination of these operating conditions requires their consideration with respect to each other before final decision as to the type of cylinder oil to use. For example, pressure alone will usually be of minor concern but pressure plus temperature and velocity have a decided relation to the viscosity of the oil and the readiness with which it will atomize when applied by the prevailing lubricating equipment. In other words, the higher the temperature and velocity the greater will the oil be reduced in viscosity and the more easily will it atomize when fed into the steam line. Furthermore, the resultant film on the cylinder walls will be more fluid. So the initial viscosity of the oil must be sufficiently high to obviate this condition, otherwise impaired lubrication may result.

Moisture, in turn, will affect the amount of compound as already noted. From the viewpoint of composition of the oil, this item is usually regarded as of greatest importance. Steam as it is used in many plants will be saturated and usually contains a few per cent of moisture; hence, the need for compounded oil to protect the cylinder walls, rings and valve mechanisms of the various reciprocating pumps and engines in which it is used. As already noted, the presence of compound in cylinder oils may result in carry-over with the condensate; therefore, greater care is required in their application to guard against excessive usage. On the other hand, any cylinder oil which contains compounds of animal fats will atomize more readily than a straight mineral oil, so it can be applied in a more varied manner, a fact which is often of decided advantage to the power plant. Obviously, any such advantage should not be availed of to the detriment of the process. All must be considered when making original selection of the oil and deciding upon its characteristics.

To determine the degree of satisfaction which is being obtained from a steam cylinder oil, one must observe the condition of the contact surfaces of all reciprocating steam machinery being lubricated; assuming that there is no objection to the amount of compound, or to its presence in any condensate used elsewhere in the plant in open heaters. Testing is not usually necessary. Instead, if the cylinder walls and valve surfaces are inspected when time permits removal of cylinder heads or valve chest covers, the nature of the oil film can be readily noted, provided such inspections are made while the surfaces are comparatively warm. That is, the heads or covers should be removed immediately after shutting down. If the contact surfaces appear rough dry, rusted or dull, they are probably being imperfectly lubricated. Conversely, if pools of oil are noticed in the counterbores, too much oil is being used. Adequate lubrication is indicated by a uniformly thin film of oil and a polished surface below.

The Oil Separator

An oil separator becomes most effective where the designer has considered the way in which cylinder oil will be carried by the steam to be purified. There is a theory that it will appear both in the form of an emulsion (due to the compound content) and as suspended mist. This latter is attributed to the straight mineral component. As a rule, scarcely any free oil can actually be seen unless an excess has been used; the steam will usually have a milky or opalescent appearance, or look somewhat like water which contains a large amount of air.

It is not a difficult matter for a well designed oil separator to break up these mixtures and to remove a large part of the entrained oil from exhaust steam. Some interesting analyses have been run to show the extremely low oil content of the purified product; they indicate that such methods of purification are a definite adjunct to a high rate of heat exchange efficiency.

Knowing that there is a definite amount of heating value to exhaust steam, it is contrary to our present ideas of economy to pass this into the air. Hence, the modern power or industrial plant engineer endeavors to put it to work in order to reduce his fuel bills. This is quite practicable if he understands the principles of steam cylinder lubrication and works toward obtaining as pure an exhaust as possible. Frequently steam cylinders are over lubricated, and this must be corrected. Oils may have been carelessly selected due to price considerations, in which case one should give more study to this matter the next time the lubricating engineer calls, and attention given to the various means for oil separation. It is a phase of power plant operation where cooperative study becomes advantageous.

Bituminous Coal Industry Considers Research

At a time when the bituminous coal industry is struggling with such immediate problems as price stabilization, sale of slack sizes of coal and maintaining a sound financial condition, one might expect that there would be no reserve energy left inside of the industry with which to plan for the future, that is, for 1940 and the succeeding years. Leaders in the industry know, however, that progress will not wait, that the future must be planned today, and that only scientific procedure can give them the basic information on which to base their plans.

Behind them in their planning are the experiences of a successful three-and-one-half year program of research on practical, technical problems of the industry that have been completed by Bituminous Coal Research, Inc. Founded in 1933 "to encourage, foster and promote. . . the utilization of coal and its products. . .by means of study, research, educational and other effort. . .", the corporation was organized to pool funds from a scattered industry for cooperative attacks on barriers that beset its way to progress and profit.

Many men in the coal industry know that BCR has accomplished much good work and that its research findings are of value all out of proportion to the expenditure of money and personal effort. It is also generally admitted that for an industry as large as the bituminous coal industry, the size of the BCR program and budget were hardly adequate to the needs.

Resolutions at Pittsburgh Meeting

Therefore, a group of executives from the coal-producing and coal sales companies, from the railroads and from companies that manufacture supplies for the coal industry met in Pittsburgh on January 7 to recommend ways and means for increasing the effectiveness of Bituminous Coal Research, Inc. Their various suggestions were embodied in a resolution to the Board of Directors of the corporation, from which the following excerpts are taken:

"Whereas, It is recognized that the application of research and engineering is essential to retain and develop coal markets. . . and

"Whereas, . . . Bituminous Coal Research, Inc. has in the past three-and-one-half years conducted a notably successful program of research on certain phases of preparation, utilization and (on) development of new markets for coal:

"Be It Resolved and Recommended,

"1. That BCR be continued and that the size of its program of coal research at established institutions be increased.

"3. That a campaign be instituted to acquaint the bituminous coal industry and associated industries with BCR, its purpose, its accomplishments, its program, and its need for funds.

"4. That funds for the sponsorship of research on bituminous coal be solicited from coal-producing companies, coal sales companies, coal trade associations, coal-hauling railroads, suppliers of materials, equipment, and electric power for mining and preparation of bituminous coal, and from mine labor organizations. . .

"9. That representation on advisory boards of BCR be given to groups of contributors as well as to engineers and scientists who can assist with technical features of the programs."

An appropriation of $235,000 per year was suggested, the money to be requested on the bases of one mill per ton of coal shipped and one-half mill per dollar of sales to the bituminous coal industry.

The action of the group that met in Pittsburgh does not of itself prove that the industry as a whole has awakened to the possibilities of research. The sustaining fact lies in the general attitude of leading coal executives and engineers in many companies who have expressed themselves heartily in favor of BCR and its purpose as well as endorsing the coal research at independent institutions. Several companies have already pledged one mill per ton for a three-year period as a concrete expression of their supporting interest.

The obvious obstacles in obtaining the deserved support from coal producers—the lack of reserve funds and a shifting status of price stabilization efforts—can be met squarely by the positive statement that so long as the bituminous coal industry hopes to continue to do business, it must either pay for research and benefit by it, or pay the infinitely greater cost of not progressing and profiting with research. Fortunately, many practical problems of the industry, similar to those already investigated by BCR, can be solved and reported in a period of one year or less after the initiation of the studies. Such researches can rightfully be paid for out of operating budgets of the contributing companies. Coal research can be designed to help the coal industry out of the financial red.

Many Research Problems Undertaken and Reports Available

The type of work done by BCR, several of whose studies have been reported in COMBUSTION, has helped both the coal and the heat-power industries. Although the work has been financed in the past by only a small fraction of the coal companies, the results of the studies have been made available to all. Following is a list of the reports issued to date, most of which can be obtained from the office of the secretary located in the Southern Building, Washington, D. C.

PUBLICATIONS

"Résumé of Research in Coal and Coal Utilizaton 1935"**	50 c
"Résumé of Research in Coal and Coal Utilizaton 1936"	50 c
Technical Report No. 1, Part I—"The Relation of the Size of Bituminous Coals to Their Performance on Small Underfeed Stokers"**	25 c
Technical Report No. 1, Part II—"Burning Tests on Four Typical Coals." This publication includes a résumé of Part I—"The Relation of the Size in the Hopper to that Burned in the Retort and Laboratory Equipment and Method of Testing"	50 c
Technical Report No. 2—"Progress Report on Coal Hydrogenation"	25 c
Technical Report No. 3—"Efficiencies and Costs of Various Fuels in Domestic Heating"*	40 c
Technical Report No. 4—"Fundamentals of Combustion in Small Underfeed Stokers"	50 c
Information Bulletin No. 1—"Clinker Formation in Small Underfeed Stokers"	10 c
Information Bulletin No. 2—"Combustion of Bituminous Coal in Small Underfeed Stokers"	25 c
Information Bulletin No. 3—"The Use of Bituminous Coal in the Dehydration of Alfalfa and Other Forage Crops"	(Free)
"The Performance of Several Types of Bituminous Coal in Small Underfeed Stokers"	20 c
"Status of Our Knowledge of Coal Selection for Steam Generating Equipment"	(Free)
"Segregation of Coal in an Industrial Steam Plant Bunker"*	(Free)
"An Experimental Investigation of the Use of Oil for the Treatment of Coal"*	(Free)
"Performance of Automatic Air Control for Small Underfeed Stokers"	(Free)
"Automatic Residential Heating with Bituminous Coal"*	(Free)
"Relation of Characteristics of Bituminous Coals to Their Use in Pulverized Firing"	(Free)

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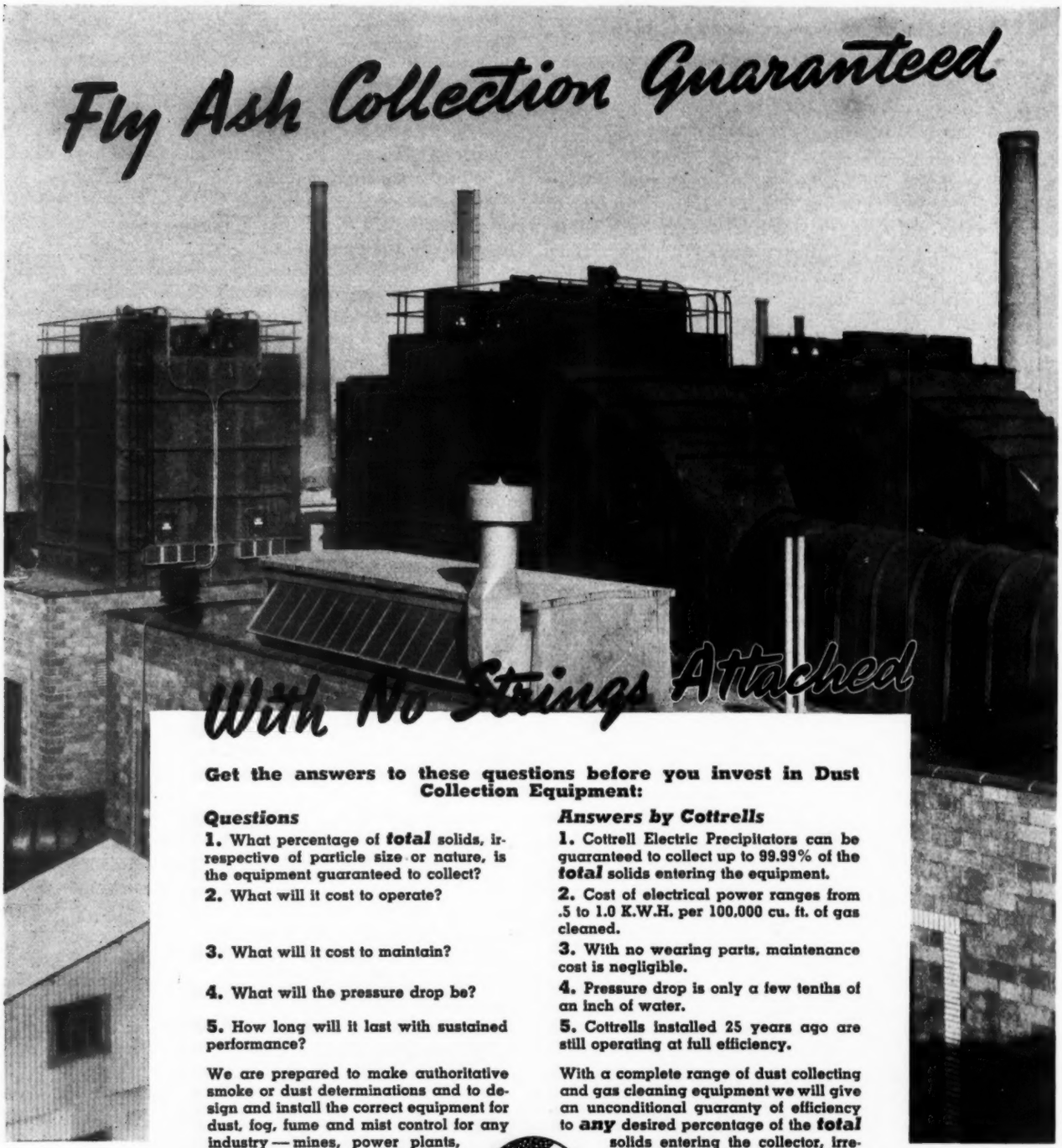

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PIONEERS IN DUST AND FUME CONTROL

Program of A. S. M. E. at New Orleans

Further information is now available as to the titles of papers and authors for the Spring Meeting of the A. S. M. E. at New Orleans, February 23 to 25, a general reference to which appeared in our January issue. In so far as concerns papers of particular interest to power engineers the program is as follows:

Thursday, February 23

9:30 a.m.

BOILER FEEDWATER STUDIES

Grooving of Tubes in High-Pressure Boilers, by R. E. Hall and E. P. Partridge
Mechanical Purification of Steam Within the Boiler Drum, by M. D. Baker
Corrosion in Partially Dry Steam Generating Tubes, by F. G. Straub and E. E. Nelson

12:30 p.m.

LUNCHEON, JOINTLY WITH S. P. E. E.

Address on Engineering Education

2:00 p.m.

SYMPOSIUM ON WASTE FUEL

Burning Oat Hulls, by C. J. Herbeck
Burning of Pulp-Mill Waste from a Sulphite Mill, by Grover Keeth
Burning Various Types of Oil-Refinery Fuels, by A. L. Wilson
Combustion of Wood-Waste Fuels, by Henry Kreisinger
Burning Bagasse, by E. W. Kerr
By-product Fuels in the Steel Industry, by Gordon Fox and W. B. Clemmitt

Friday, February 24

9:30 a.m.

POWER

Economics of Power Plants for Joint Utility and Industrial Service, by A. C. Klein and T. E. Crossan
Development of Power and Steam for Oil-Refinery Service, by C. E. Steinbeck
Mountain Creek Steam-Electric Station, by Harry R. Pearson

Saturday, February 25

9:30 a.m.

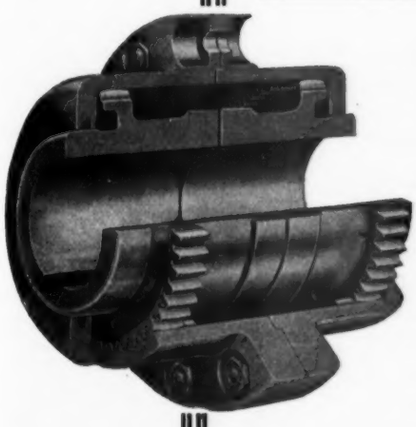
FUELS

Lignite—A Grate-Fired Fuel of the Future in the Southwest, by C. J. Eckhardt, Jr.
Pulverized Coal-Tar Pitch as a Fuel, by E. W. Clarke

Saturday afternoon is reserved for plant inspection trips and in the evening there will be a dinner dance. Headquarters will be at the St. Charles Hotel and at the same time and place the Louisiana Engineering Society will hold its Annual Meeting with several joint sessions being held with the A. S. M. E.

In connection with the meeting an official tour is planned. Groups from various cities in the North and Middle West will meet at Mammoth Cave, Ky., on Sunday, February 19. The following day will be spent at Birmingham, Ala., where pipe-manufacturing plants will be visited and social events enjoyed. From here the party will proceed to New Orleans. Tuesday will be the final day of the Mardi gras. On the return trip members and guests will have the option of returning by rail or by boat, taking in a Caribbean cruise.

POOLE FLEXIBLE COUPLINGS



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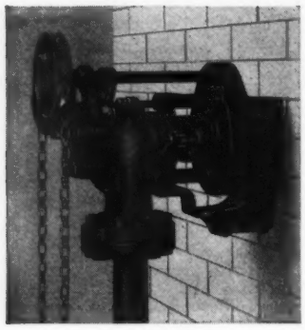

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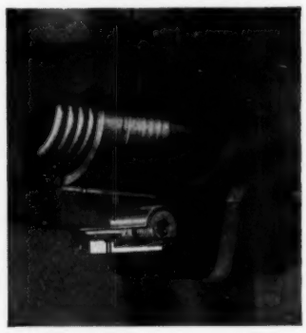

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STEAM ENGINEERING ABROAD

As reported in the foreign technical press

Some Large British Turbines

Present British turbine practice, as indicated by a description of installations recently completed or on order, is contained in the January issue of *Engineering and Boiler House Review* (London). Despite the fact that for central station service units of around 50,000 kw and moderate pressures predominate, there is a noticeable tendency to depart from 600 lb and 850 F, which has become more or less common practice during some few years past, and to go to both higher pressures and temperatures; also, in some cases larger units.

There is an increasing demand for generation at 33,000 volts, and two-cylinder tandem machines with double exhaust are employed in the majority of cases. Considerable investigation has been carried on in connection with vibration of wheels and blades, turbulence effects and gland leakage. One turbine builder has lately introduced the vortex type of blading for low-pressure stages.

The following is briefed from the descriptions of some of the recent units:

The highest pressure in England is represented by the extension to Brimsdown Station where a 19,000-kw topping turbine-generator receives steam at 1900 lb 930 F from two 210,000-lb per hr Loeffler boilers and exhausts to a 31,000-kw low-pressure machine. There are now on order two more Loeffler boilers of the same pressure and temperature, but of 250,000 lb per hr capacity, which will supply a 60,000-kw two-cylinder machine consisting of a 19,500-kva high-pressure element and a 43,600-kva low-pressure element.

A 100,000-kw turbine-generator, consisting of a 16,000-kw, 3000-rpm high-pressure element taking steam at 1350 lb 950 F and exhausting at 600 lb to a 78,000-kw two-cylinder low-pressure element, as well as a 6000-kw house turbine, is now being installed at Battersea Station.

At Iron Bridge Station a third 50,000-kw unit has lately been completed and a fourth is on order. These last two are two-cylinder machines with 19 high-pressure stages and 8 double-flow low-pressure stages. The steam conditions are 375 lb 800 F and the generator voltage 33,000.

Barking B Station has four 75,000-kw, 600-lb, 850 F units.

A second 51,600-kw unit has recently gone into the Kearsley Station in Lancashire and one of like capacity was placed in service last May at Barton Station. The latter is a two-cylinder 1500-rpm machine having 25 high-pressure impulse stages and 16 low-pressure stages with double exhaust.

Clarence Dock Station (Liverpool) has four 50,000-kw, 600-lb, 800 F units running at 1500 rpm and having 28 impulse stages in the high-pressure element and 16 low-pressure stages with double exhaust.

At Blackburn Meadows (Sheffield) one 50,000-kw, 1500-rpm, two-cylinder machine has been ordered to operate at 570 lb 825 F. This is a duplicate of that installed in 1937. This station also contains a 33,000-kw, 3000-rpm, three-cylinder impulse-reaction turbine, supplementing one of 25,000 kw placed in service in 1932 and another of 30,000 kw in 1935.

Dalmarnock Station in Glasgow has installed two 50,000-kw, 1500-rpm machines, operating at 600 lb 825 F. These are of the two-cylinder pure reaction type and employ four stages of extraction.

A considerable number of other units of smaller capacity were listed, including one of 30,000 kw for the Thorpe Station (Norwich) which has 48 high-pressure stages and 7 low-pressure stages, with steam conditions of 650 lb 800 F.

Italy's Natural Steam Plants

Italy, although possessing considerable water power, is very poor in fuels supply. Her desire to be independent of the importation of foreign fuels for power generation has led to renewed interest in the natural steam seams that underlie the district of Toscana and the high valleys of Cecina and Cornia, embracing an area of about 75 sq mi. The steam supply from these seams is constant and the pressure uniform, ranging from 64 to 71 lb per sq.in. in different sections, depending upon the depth of the drillings. It has entrained about 5 per cent carbonic acid and minor percentages of other substances, including boric acid.

The first recorded use of this steam was in 1818 when water from the adjacent lagoon was evaporated by the steam to obtain boric acid.

Its first application to power generation dates back to 1904 when the steam was put through a small reciprocating engine to provide electricity for a few lamps. A year later a larger engine of about 40 hp was installed and remained in service for 15 years. A small central station containing a 250-kw condensing turbine-generator was erected in 1912. Steam for this turbine was obtained from evaporators heated by the natural steam, as it was feared at that time that, if used direct, the boric acid content of the ground steam might cause trouble and the entrained gases would involve large and costly air pumps for the condenser.

Despite this, there were later installed three 3000-kw Parsons turbines using the natural steam direct after it had passed through apparatus for removing the condensible gases. These units were relatively large in physical dimensions because of the low initial pressure. The equipment for removing the gases is represented by the accompanying sketch and consists of two large drums, one horizontal and the other vertical. The former contains water through which the ground steam in the central perforated tube is bubbled. The con-

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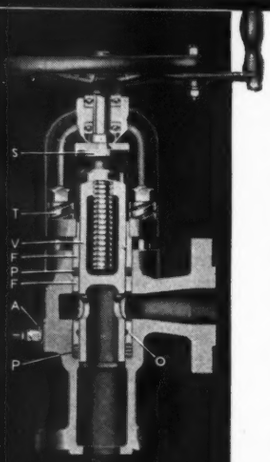

Yarway Seatless Blow-off Valve. Operation: After Valve is closed, shoulder S on plunger V contacts with upper follower gland F, forcing it down into body and compressing packing P above and below port. Annular groove O connects with Alemite fitting A for lubricating plunger and packing. Yoke springs T maintain continuous pressure through follower gland F on packing rings P.

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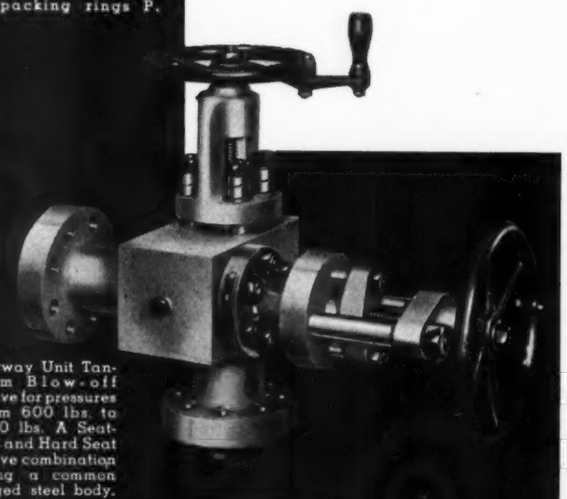
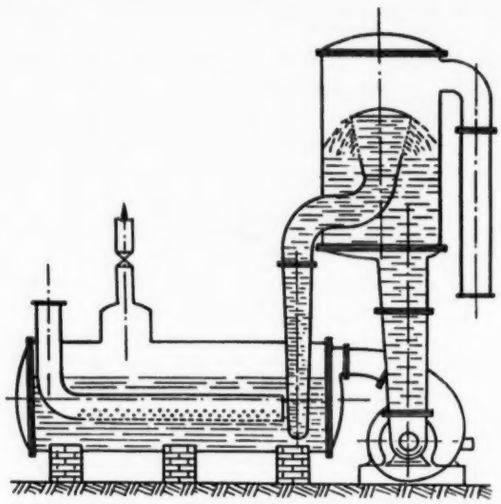


Yarway Unit Tandem Blow-off Valve for pressures from 600 lbs. to 1500 lbs. A Seatless and Hard Seat Valve combination using a common forged steel body.

YARWAY

BLOW-OFF VALVES

densible gases collect above the water and are drawn off through the dome. The steam thus cleaned is condensed by the water which becomes heated and the condensate mixture is pumped to the top of the vertical drum where it spills over the central funnel and is re-evaporated at slightly reduced pressure. The water leg of the vertical drum feeds the pump which keeps the horizontal drum supplied. The pipe at the left delivers the steam from the vertical drum to the turbine.



Means of removing condensible gases

A later installation of 750-kw turbines uses the natural steam directly without extracting the gases and the exhaust is led to chemical equipment for the recovery of boric acid, carbonic acid, ammonia and other rarer constituents such as argon, neon and helium. The exhaust still has sufficient heat to serve as the heating medium for the extraction process. In this case the relatively low thermodynamic use of the turbine is offset by the simplicity and low cost of the installation. Moreover, the cost of steam is nil.

Earlier drillings to a depth of 500 ft yielded quantities of steam up to 66,000 lb per hr, whereas recent drillings to greater depth, and of larger diameter have yielded 440,000 lb of steam at 71 lb pressure, 400 F per hr per well. The total present output of the district is 2,650,000 lb per hr.

The generating capacity now installed is 66,000 kw which produces about 500 million kilowatt-hours annually. It is estimated that approximately a tenth of the present total energy requirements of Italy could be provided by ground steam from the Toscana district.

The description from which the foregoing is abstracted appears in *Archiv für Wärmewirtschaft und Dampfkesselwesen* of August 1938.

La Mont Boiler Tests

Results of trials on the Thompson-La Mont boiler recently installed at the works of G. & J. Weir, Ltd., are given in the January issue of *The Power and Works Engineer* (London). A brief description of this unit appeared in the December 1938 issue of COMBUSTION.

While the unit was designed for a pressure of 1000 lb per sq in., it will be noted from the table of results that the test was run at a little over 900 lb. It is explained

that this was done as a matter of practical operating expediency and that no difficulty was experienced in operating at the full 1000 lb. At present the unit is operating at an extremely variable load and under these conditions if run at 1000 lb with little margin below blowoff pressure, the safety valves are likely to lift. Moreover, the turbine was designed for 850 lb. Following is a digest of the test performance:

Evaporation, lb per hr	40,748
Steam pressure, lb per sq in	907
Steam temperature, deg F	877
Boiler exit gas temperature, deg F	766
Economizer gas outlet temperature, deg F	296
CO_2 at economizer outlet, per cent	15
Gross heating value of fuel as fired, Btu per lb	10,865
Moisture, per cent	16.77
Overall efficiency of unit, per cent (gross)	86.46
Heat lost in dry flue gas, per cent	4.82
Heat lost by moisture in flue gas, per cent	5.5
Heat lost through incomplete combustion, per cent	0.39
Heat lost through combustible in ash, per cent	0.736
Unaccounted for and radiation losses, per cent	2.094
Power absorbed by auxiliaries, per cent	1.25
Net overall efficiency, per cent	85.21
Power absorbed by circulating pump, per cent	0.55
Heat release in furnace, Btu per cu ft per hr	39,760
Guaranteed overall efficiency, per cent	81
Excess of guaranteed efficiency, per cent	5.46

It will be noted that despite the small tubes and their nozzle restrictions to distribute circulation, the power consumed by the circulating pump was only 0.55 per cent.

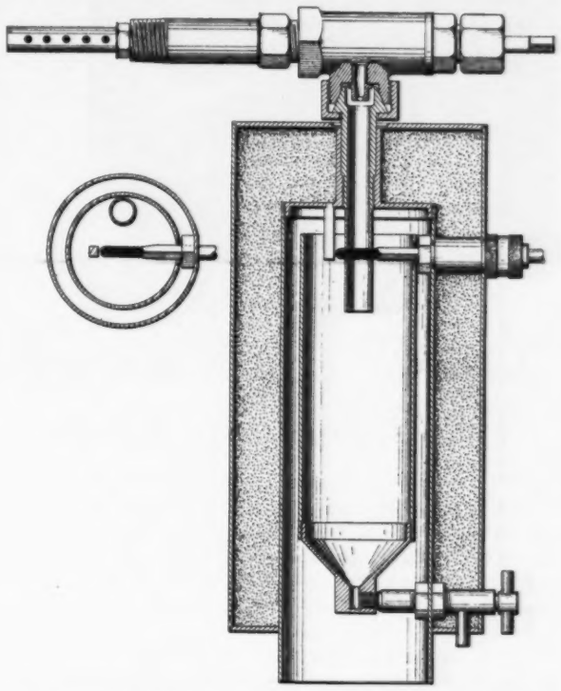
Creep and Recovery at High Temperatures

The Brown Boveri Review for December 1938 devotes considerable space to a report from that company's materials-testing laboratory on creep and recovery at high temperatures. This discusses the influence of alloying, heat treatment and deformation on the creep limit; the extent to which the evaluation of the creep-test results depend upon the method employed; the unreliability of extrapolation over a very long period; creep tests with interspersed periods of relief, under constant and under falling temperature; and an explanation of the recovery by consideration of the mechanism of deformation of the grain boundries and crystals.

With reference to alloying, if a high creep limit is to be attained, the importance of molybdenum is generally recognized. For temperatures of 750 F to 840 F, 0.3 to 0.8 per cent molybdenum is widely used for turbine shells, and for temperatures over 900 F chromium-molybdenum or chromium-nickel-molybdenum steels afford a very high creep limit as well as high strength when heat treated. The prediction is made that 930 to 1020 F will be the limit for low-alloyed steels, above which the resistance to deformation and to scaling must be reinforced by more effective means, that is, by austenitic steels. The addition of tungsten, by the formation of strongly temper-resisting tungsten carbide, helps to improve austenitic steels. However, while the addition of molybdenum considerably strengthens the creep limit of pearlitic and martensitic steels it does not act in this manner to any marked degree on austenitic steels.

Aside from improving creep values by alloying, experience has shown that, assuming correct alloying, much can be done by heat treatment. At what temperature the effect of heat treatment ceases under constant heating is determined by the temper resistance of the heat-treated structure and by its carbide content, but temperatures of 930 to 1020 F are such that a pronounced increase in creep limit by proper heat treatment results.

ELLISON U PATH STEAM CALORIMETER



Accuracy within 2°F of theoretical temperature

STEAM PATH: Steam enters and escapes at the top of the steam chamber, flowing in a U path, then flows down through the steam jacket and escapes at the bottom. Evaporation of moisture momentarily in excess of the throttling process is separated in the chamber and is re-evaporated and accounted for on the thermometer when the normal superheat in the chamber is restored, forming a combined throttling, separating and re-evaporating calorimeter in a single chamber.

INSULATION: The steam chamber is insulated in a steam jacket with steam of the same temperature as the expanded steam in the chamber, 1″ of lamp black and a nickel plated casing.

"An Ellison calorimeter installed with the valve handle removed and the uninsulated metal parts covered with many layers of tissue paper, gave an expanded steam temperature of 322°. This temperature was obtained using the standard sampling nipple supplied with the calorimeter and with the load on the boiler reduced to approximately 50% of rating. The theoretical temperature corresponding to these conditions as closely as we can determine it from the A.S.M.E. data published in the February 1926 issue of "Mechanical Engineering," is 324° plus or minus .2°. This we feel is a very remarkable agreement and one which demonstrates the thorough adiabatic construction of the calorimeter."

H. G. Smith, Hagan Corporation

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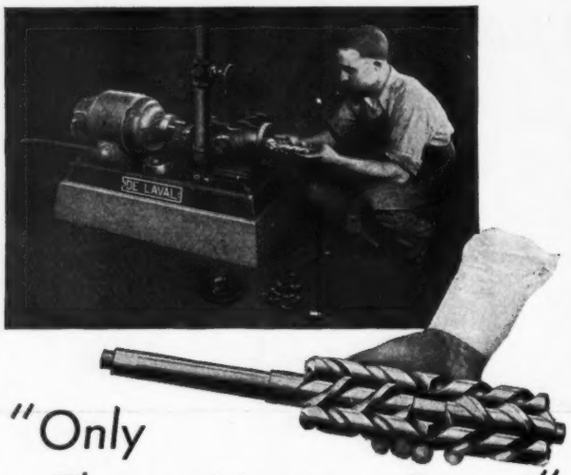

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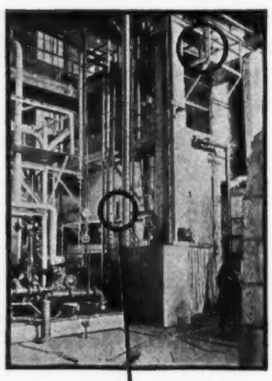

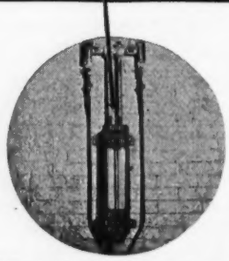

Reliance EYE-HYE at International Nickel Co. power plant, Huntington, W. Va., easily read from operating floor.

Above—Reliance Water Column and Gage.

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The non-existence of a uniform method of determining creep limit is a serious difficulty. A test duration of 1000 hr is generally advocated by British and American investigators, whereas those on the Continent advocate the shorter test duration of 40 to 100 hr. Inasmuch as very long-time tests cannot be employed in practical engineering, the evaluation must take into account the manner in which the creep results are arrived at. The somewhat greater unreliability of the short-time test is compensated for by using only two-thirds of the creep limit as the allowable stress. At this admissible stress, the creep rate is not only much lower but the overload capacity of the steel is much better. Nevertheless, the long-time test of 1000 hr is indispensable for making studies and gathering information.

The conditions of an ordinary creep-limit test are often affected in practice by other influences on loaded parts of the machinery. The influence of several repeated loads and unloading with pauses between at different temperatures is of interest. After the unloading, a general contraction is noted but this never attains the amount of the preceding creep elongation. When successive loading and unloading is applied the creep and creep rate diminish rapidly with each new load and similarly to an uninterrupted creep limit test. Unloading pauses at 930 F show a somewhat greater resulting elongation than those at room temperature.

The conclusion is reached that there are considerably smaller elongations, as a result of varying loading and unloading in service, than could be expected from the uninterrupted long-time creep test.

Corrosion in Fittings and Steam Lines

An article by W. Schumann in the January 1939 issue of *Archiv für Wärmewirtschaft und Dampfkesselwesen* deals with corrosion in boiler fittings and feed lines. Attention is called to the fact that where acid treatment is employed to remove carbonate hardness corrosion often takes place in the valves and feed lines. Also, attack from entrained oxygen or H_2CO_3 is more likely where the inside surfaces of the lines are rough or where the fittings are entirely free from scale. Such attacks are usually in the form of rust postules under which will be found craters. Avoidance of dead corners or pockets in the lines will lessen the opportunity for such attack.

Leaks under high pressure will cause cutting and badly closing valve seats, in which connection gray iron or cast steel is especially susceptible.

Where the water treatment leads to the accumulation of alkaline salts within the boiler, the seats of blowoff valves are often attacked where copper or zinc-copper alloys are used. These metals are not permanent in the presence of strong alkaline solutions. The metal of such seats must be tough and hard to prevent such alkaline scale from penetrating the surface during the pressing or rubbing action incurred in closing the valve. Phosphor-bronze, chrome or chrome-nickel steel is therefore preferable for such service.

Nitrided steel is recommended to resist cutting from steam impact but it is not very corrosion-resistant at high steam temperatures. For high temperatures 13–18 per cent chromium steels are better suited and many of the more recent high-pressure high-temperature valves in Germany are of such composition. It is important,

however, to guard against porosity and improper fits between the dissimilar metals.

In locations where a swirling action or disturbed flow of the steam passing through fittings is likely to occur, the metal may wear away in lanes. Streamlined fittings and highly resistant metals should be employed in such places.

Steam vs. Diesels for Marine Service

In reviewing technical progress in marine engineering during 1938, *The Shipbuilder and Marine-Engine Builder* (London) for January quotes figures from Lloyd's Register showing that motorships now constitute 22.8 cent of the total world tonnage. This is more than double that of ten years ago. The present trend in propelling machinery is indicated by figures now available for the first nine months of 1938 which, in terms of horsepower installed during that period, shows 10.6 per cent for reciprocating steam engines, 22.9 per cent for steam turbines and 66.5 per cent for diesel engines.

In Great Britain and Ireland a less predominant position has been attained by diesel engines for, of the total propelling machinery under construction, steam accounted for 50.9 per cent and internal-combustion engines for 49.1 per cent. Great Britain's output of marine diesel engines for the period mentioned was 418,231 hp, or 21.4 per cent of the world total; followed by Germany with 336,366 hp, or 17.2 per cent, and by Italy with 239,200 hp or 12.2 per cent. Coal now serves only 47.2 per cent of the world's tonnage.

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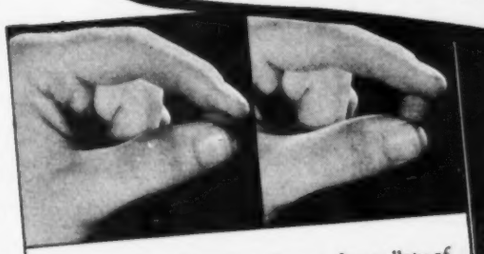

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Port Washington's Heat Rate

Data on the third year's performance of the Port Washington Station were given in the January issue of COMBUSTION. In this article the net station heat rate for the year was given as 10,803 Btu per kwhr. This was incorrect, the actual figure being slightly better, or 10,788 Btu per kwhr. The error was brought about through applying arithmetical rather than weighted averages to certain last-minute figures.

Obituaries

Dr. Alphonse A. Adler, for many years a New York consulting engineer in power plant work, died suddenly on January 13. He was a graduate of Cooper Union, later receiving his doctor's degree from New York University. In addition to his consulting practice he had taught at Brooklyn Polytechnic Institute and at the time of his death was giving a course at New York University. He had long been active in local affairs of the A.S.M.E. and the A.S.H.V.E.

N. P. Lord, production manager of the Spencer Turbine Company, died at Hartford, Conn., on January 4. He had been associated with this company since 1926.

William R. Seigle, vice president and director of research of Johns-Manville Corporation, died on December 26 following an operation at Rochester, Minn. He was 59 years old and had been associated with that company for nearly forty years.

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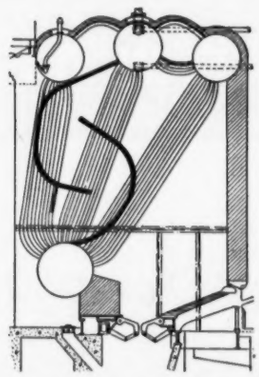

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